GEORGE F. NORDENHOLT, *Editor*

Strong Leadership Is Needed

OVER MANY CENTURIES man developed from a conglomeration of savage individuals to organized tribes and then nations. In the course of this process it was necessary for the individual to give up some of his so-called prerogatives, such as stealing, or killing his neighbor if he did not like him. In place of that the individual had to agree that it would be required to bring quarrels before appointed judges and abide by the verdicts rendered. Obviously, such a social organization can function only if its leaders are feared or respected and if the individuals generally recognize that they must subordinate their selfish desires as might be required for the good of the social whole.

Strong leadership carries with it the respect and confidence of the masses and their voluntary unstinted efforts to sacrifice for the good of the nation. Witness England amidst its blood, sweat and tears. When leadership with a strong guiding hand is lacking, national unity disappears. Then vested selfish commercial interests exploit society to fill their coffers with ill-gotten gains while labor feathers its nest without regard to the consequences to the nation. Both these groups act as if oblivious to the fact that they stand to lose far more than the pittance gained, as demonstrated by the recent collapse of France. Leadership was lacking.

To-day in the U. S. A. we can no longer laugh off the innumerable strikes that are disrupting our defense program. The "fraction of one percent" lost time is growing at an alarming rate as labor becomes more arrogant, more demanding, more domineering. Admittedly, in numerous instances labor is justified in demanding higher wages; in numerous other instances the demands are exorbitant. Some of the strikes have been over the question of who will receive the union dues collected. Often the strikers represent but a small minority of the employees. In several instances the strike was a so-called outlaw strike, probably instigated by men of the communistic or pro-Nazi type.

But labor is not entirely to blame. In a few instances the industrial corporation has been the offender. In the case of the southern coal operators, the public was informed that the representatives of the corporations involved simply would not cooperate in that they refused to attend a meeting arranged to mediate the controversy.

Obviously, strong leadership in the highest places of government will be needed, and needed quickly, to put and maintain our preparedness program on the most effective basis. We have a Department of Labor. Presumably it is the responsibility of the Department of Labor to iron out labor difficulties. In this task it has failed. Perhaps in recognition of its failure, various other commissions or bureaus have been established in a more or less fruitless effort to cope with the situation.

But the essential ingredient, leadership, is still lacking and labor has continued on its road to national chaos at an accelerating pace. It seems that the Secretary of Labor does not command the confidence of either labor or industry.

Engineers have designed and are improving the armaments; other engineers have created the equipment and organization to make possible the rapid and efficient production of the materials needed for national defense; industry as a whole is tackling its job with little or no dollars profit; hundreds of thousands of our young men are serving in the Army and Navy for a mere pittance. The only thing lacking is a strong personality in the office of Secretary of Labor, who has earned the confidence of labor, industry and the public and who can use forceful persuasion when necessary. Such a Secretary of Labor armed with proper authority would soon put the spirit of labor and industry on the plane on which it must be if democracy is to survive.

Secretary of Labor is now, above all times, a he-man's job, not an office to be occupied by a woman, regardless of the abilities of her subordinates and associates. None can deny that the labor situation borders on chaos and has already seriously interfered with the defense program. The Labor Department has failed in getting desired results. On that basis alone, in any well-organized body, political or otherwise, a change in the head of the department is in order.

Assembly line of Vultee basic trainer planes, fabricated with welded tubular steel structures

AIRCRAFT DESIGN

Factors Which Affect Production Economy

JAMES E. THOMPSON
Service and Specification Engineer, Vultee Aircraft, Inc.

SIX BASIC factors directly influence production practicability and economy: (1) simplification of design, (2) practicability of assemblies and installations, (3) elimination of close limits and clearances, (4) use of existing parts and standards, (5) weight economy, (6) serviceability.

Simplification of Design

Simplification, from the standpoint of both structural and fabrication considerations, is necessary if the design is to be practical. Therefore the design should begin with the simplest arrangement that will fulfill basic service requirements and then elaborated only enough to become a practical production design.

If the basic design lends itself to manufacture in a reasonable number of comparatively small units, each complete in itself, considerable production economies will result from sub-assembly manufacture in simple jigs. It may seem impractical, but it is the most feasible manner to obtain production economy with the complicated structures found in aircraft. With the basic structure broken down into a number of units, fabrication of extremely large units, with their attendant massive assembly jigs, handling and final assembly problems can be avoided. It is obvious that certain units, such as a wing center section, must of necessity form large units when completed—but this need not prevent the designer from planning the center section structure for fabrication from, say, three basic sub-assemblies of nose-section torsion box, intermediate section, and trailing edge section, to be joined in a simple riveting operation in a final assembly jig.

If the designer "follows through" in his mind the actual fabrication of each part, he can anticipate possible manufacturing problems and correct the design to minimize the operations involved. Of paramount importance is designing for "machinability." This factor cannot be overemphasized, as it is only too easy to complicate the machining of a part unnecessarily by lack of forethought during the design. If the machining operations required on each part are visualized, difficult operations can generally be avoided.

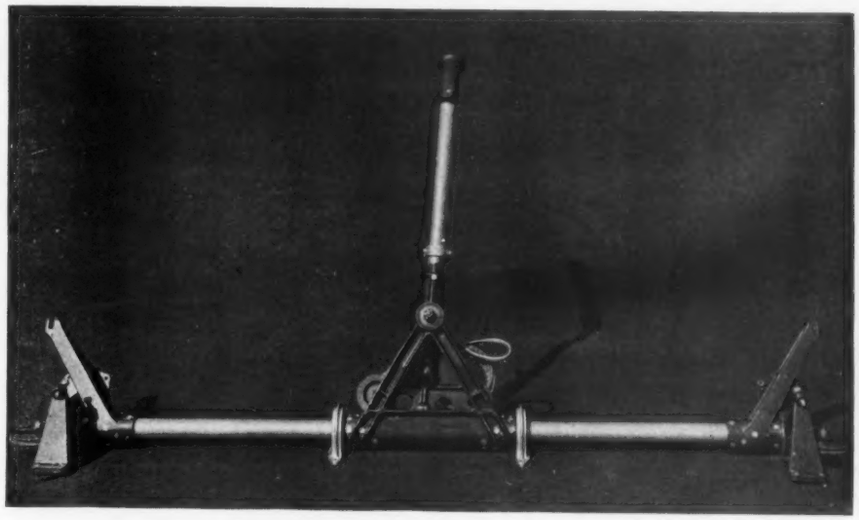
Design changes during manufacture to suit production requirements can be largely eliminated by sufficient preliminary study of the fabrication processes involved as compared with fabrication processes available in the plant. Every assembly should be simplified by minimizing the number of required fittings, combining parts wherever possible, eliminating unnecessary pieces, and substituting spot welding for riveting on secondary structures.

A good design is one which requires few changes while being translated into practice; a design which requires frequent changes is not only a cause of continual annoyance and expense, but its designer risks the charge of being impractical.

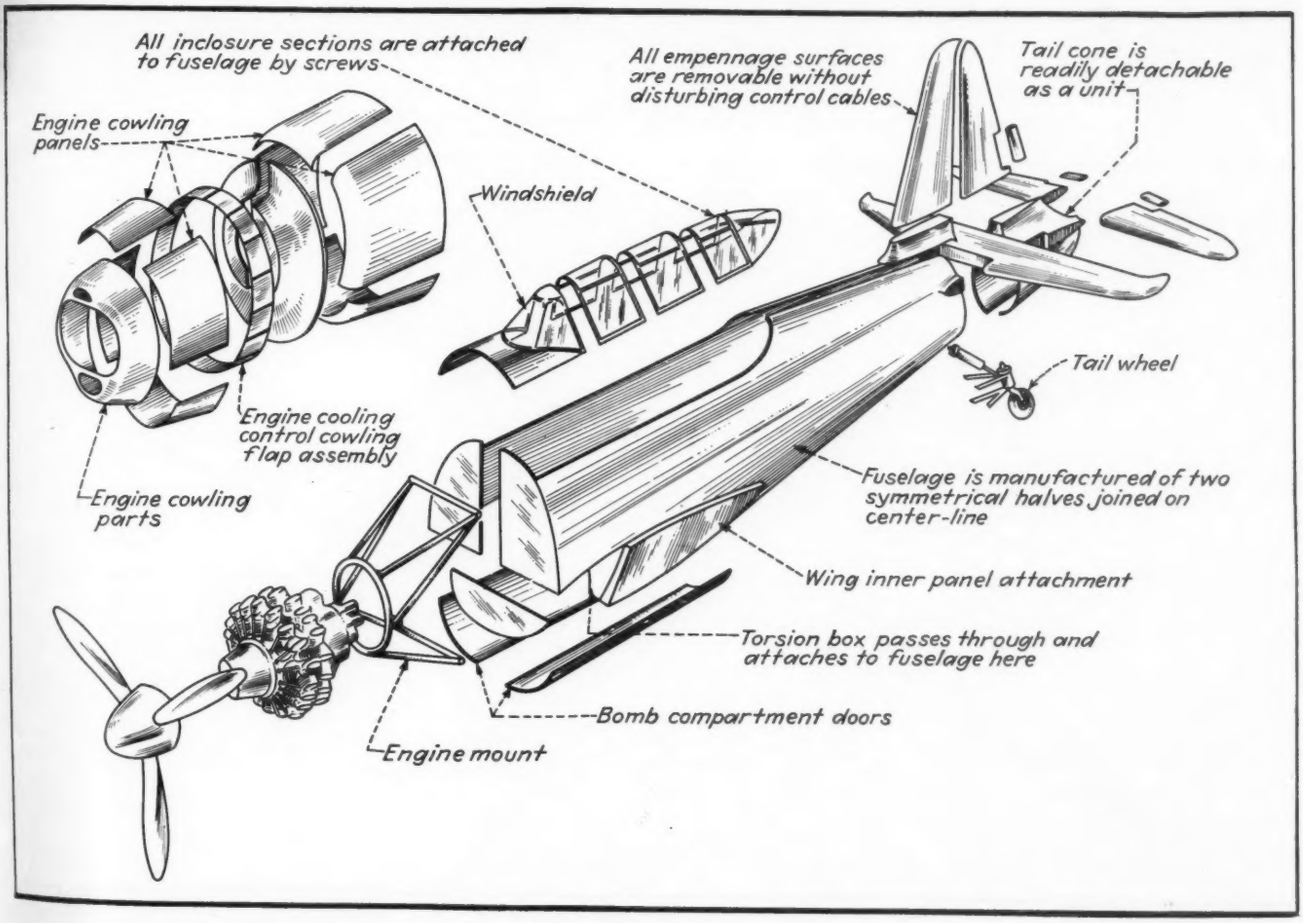
Complicated mechanisms requiring involved or delicate units and fine adjustments generally give more or less difficulty throughout production and subsequent service life. Adequate preliminary investigation will usually reveal a method of accomplishing the desired end by fairly simple means and will avoid expensive changes.

Practicability of Assemblies

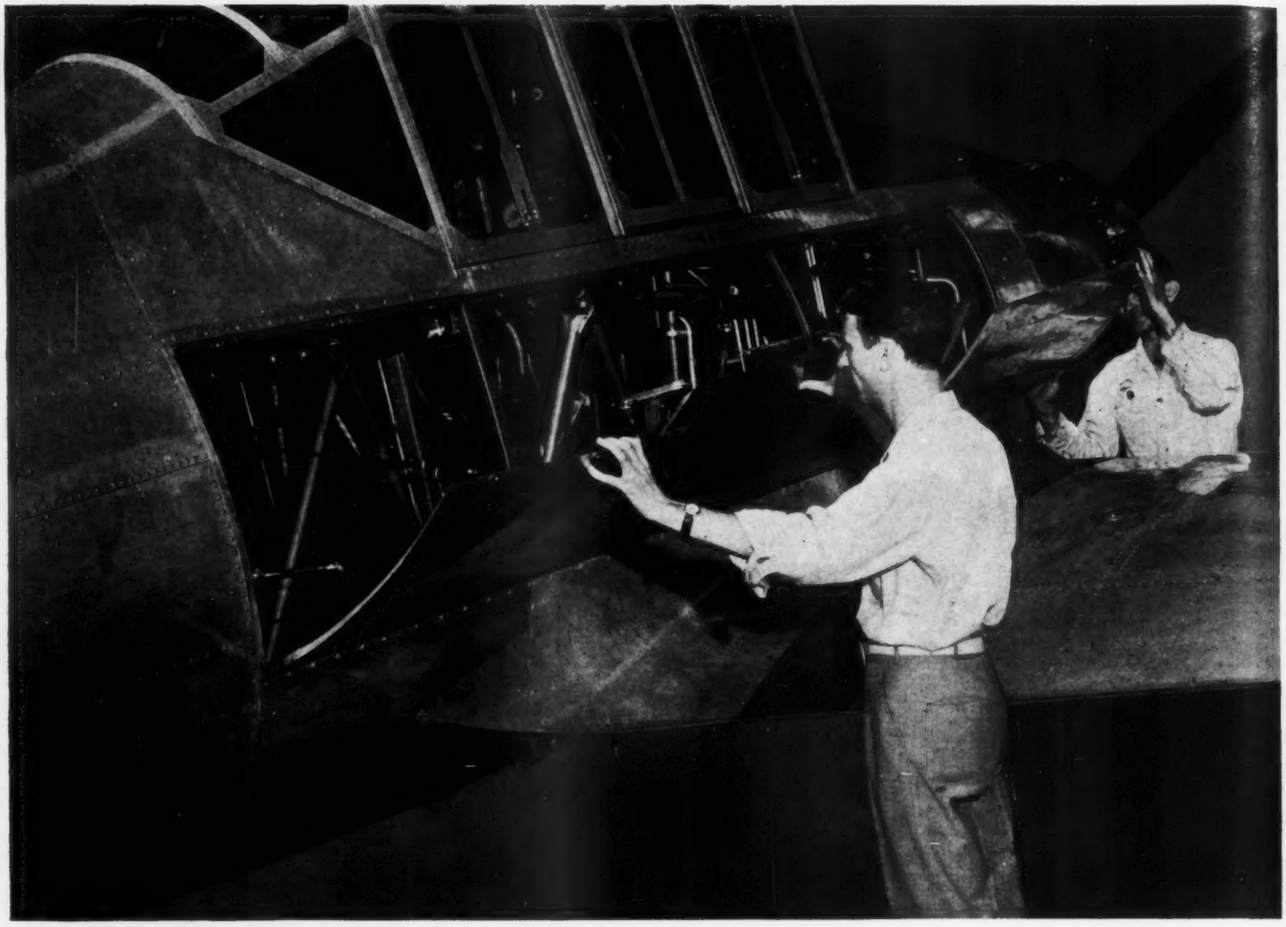
Complicated operations can be eliminated if the sequence of assembly and installation is carefully analyzed. The fact that it is possible to assemble or



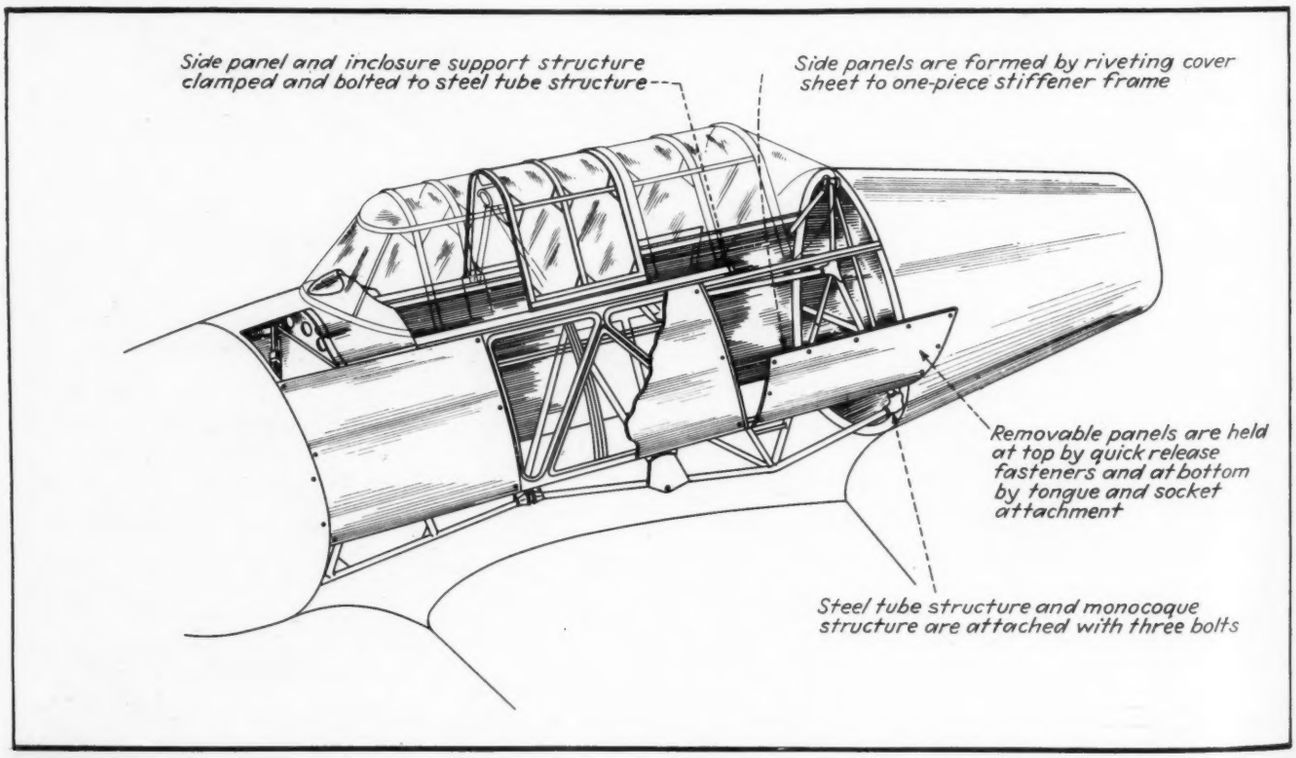
Bench-assembled control stick for pilot


All inclosure sections are attached to fuselage by screws
All empennage surfaces are removable without disturbing control cables
Tail cone is readily detachable as a unit
Engine cowling panels
Windshield
Tail wheel
Engine cooling control cowling flap assembly
Engine cowling parts
Fuselage is manufactured of two symmetrical halves joined on center-line
Wing inner panel attachment
Torsion box passes through and attaches to fuselage here
Bomb compartment doors
Engine mount

Semi-monocoque fuselage for a single-engine bomber, arranged for practical sub-assembly



Removable fuselage side panels provide complete access to all controls and fuselage equipment in the Vultee Valiant trainer. The plane has a high degree of accessibility because of its composite steel-tube and semi-monocoque fuselage


Side panel and inclosure support structure clamped and bolted to steel tube structure
Side panels are formed by riveting cover sheet to one-piece stiffener frame
Removable panels are held at top by quick release fasteners and at bottom by tongue and socket attachment
Steel tube structure and monocoque structure are attached with three bolts

Elements of composite-type fuselage is an excellent design for mass production of small single-engine planes

install a part or group of parts is not enough—the design must be such that these operations can be accomplished simply and readily, so that production time may be maintained at a minimum.

The practice of cutting, trimming, or similar operations during assembly always entails expense, as well as making replacement difficult and costly. Installations should consist of a group of parts or assemblies which can be installed at a station in the final assembly production line without fitting, riveting, threading, welding, or any other fabricating operations. It should be possible to complete all installations by non-fabricating operations such as bolting, clamping, pressing, and the like. Sub-assembly work should never be made a part of an installation, excepting the "installation" of welded and riveting brackets, if any semblance of a production line is to be maintained.

The method used to attach an assembly or installation to the surrounding structure is generally influenced by design requirements, but the designer must provide a maximum of accessibility for installation or replacement operations. Any design which requires the work to be accomplished in cramped, difficult places is fundamentally unsound, and greatly increases production costs. If the operations to complete an assembly or installation are difficult or awkward, the design should be corrected.

Bench assemblies and sub-assemblies which permit installing completed units at final assembly, rather than a mass of individual parts, are of great assistance in accelerating production and reducing costs and are particularly desirable for eliminating assembly difficulties in confined areas. Care must be used, however, to avoid unnecessarily large, clumsy sub-assemblies which are awkward to fabricate and handle at their next assembly.

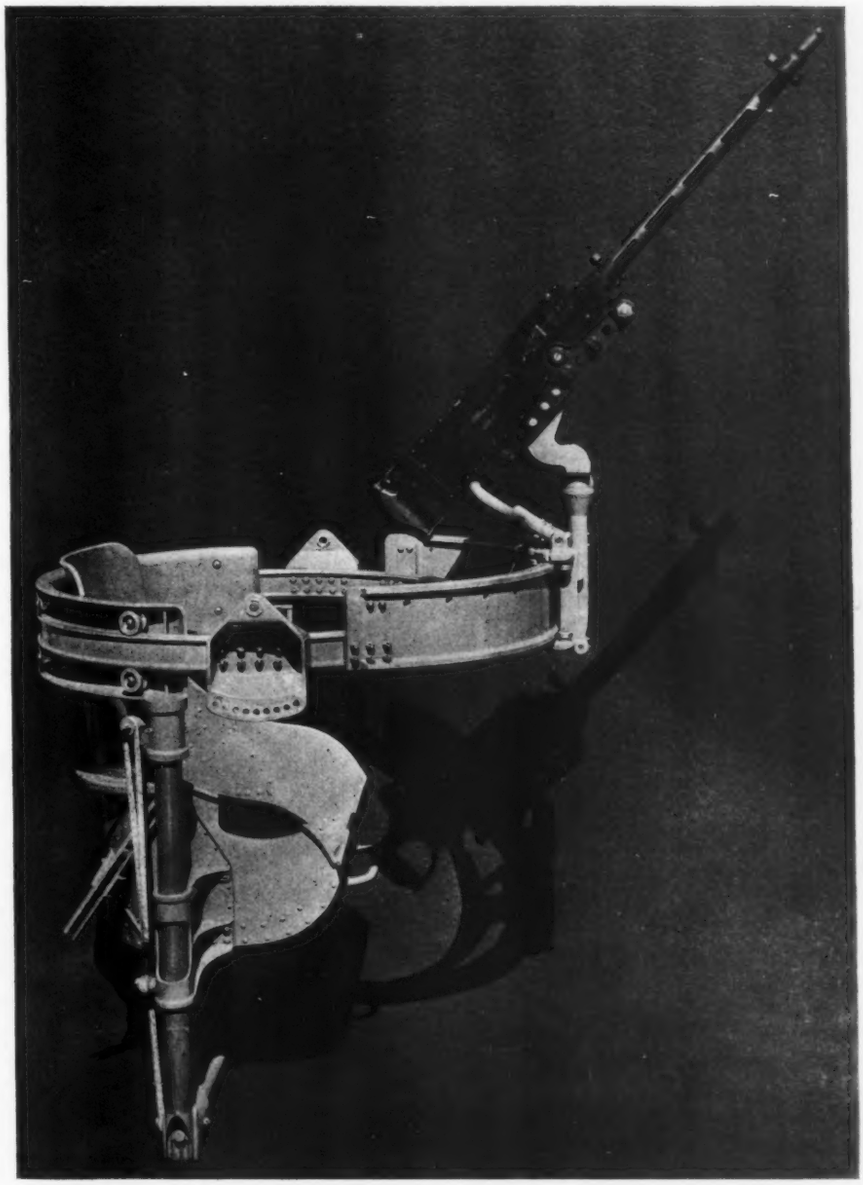
Eliminating Close Limits

Before selecting dimensional limits, the designer should ask himself the question, "Would I scrap a piece costing as much as this does, if it came from the shop 0.002, 0.003, or even 0.005 in. beyond the limits I have specified?" If these limits are tolerable, he should add that much to the limits specified on the drawing.

Always maintain limits as generous as practicable, as nothing complicates production more than the necessity of working to unnecessarily close values. The limits should favor possible error when the design permits; *i.e.*, on a turned shaft, a minus limit (thus +0.000, −0.002); on a bored or drilled hole, a plus limit (thus +0.002, −0.000); on a close-ground diameter turned shaft, a balanced limit (thus ±0.0005). A good knowledge of practical limits for high-grade interchangeable parts is helpful to the designer in avoiding designs which require impractical values.

Clearances provided for installation, assembly, and operation of structure and mechanisms, termed as "atmospheric" clearances, should always be maximum, rather than the minimum, particularly with non-related parts which must have clearance, as in a landing gear actuating cylinder which passes a structural member. In such cases, manufacturing limits on each assembly may accumulate in the negative direction, rather than balancing, with the result that a "paper" or layout clearance $\frac{1}{16}$ in. may easily become $\frac{1}{16}$ in. interference in the actual installation. Many engineering changes on new models result from this cause, for a vast number of close "paper" clearances disappear during assembly with the result that reworks and redesigns are necessary. Most of these inaccuracies are easily avoided by allowing greater clearances in the design layout. This usually is possible. The unfortunate feature, from a production standpoint, is that a great number of these cases must be reworked to such close clearances that normal assembly, installation and inspection operations are practically impossible.

Wrench clearances should receive



Bench-assembled gunner's seat, which can be installed by simply inserting two pivot bolts, thus greatly speeding production

special consideration; nothing is more annoying and time-consuming than to install a part which requires tedious tightening of nuts and bolts with a special 12-point wrench, ½ of a turn at a time. Often the part can just as well be provided with free access for a standard box-wrench or speed-wrench.

Using Existing Parts and Standards

If careful attention is given to existing parts and standards wherever possible, the use of special parts of intricate design can usually be avoided. During the preliminary investigation, make a careful study of all similar company parts and of existing commercial and government standard parts and shapes.

In many cases it may be impractical to use an existing part exactly as is but with minor changes it may be entirely satisfactory. This is often advantageous as it usually permits the use of many existing tools, templates, dies, and jigs. However, this does not mean the reworking of existing parts; rather the design of similar parts using the same tools.

The general efficiency of a design must never be sacrificed in order to use a few existing parts, for the matter of interchangeability can easily become a chimera if it is pursued to such an extreme that weight economy or the functioning of the design are affected. To be truly interchangeable, parts must be usable in all models affected, without rework or alteration. If such work is required, it is best to forget about interchangeability between models, and to design similar parts for the various models, paying attention only to obtaining some interchangeability of tools, jigs, dies and templates.

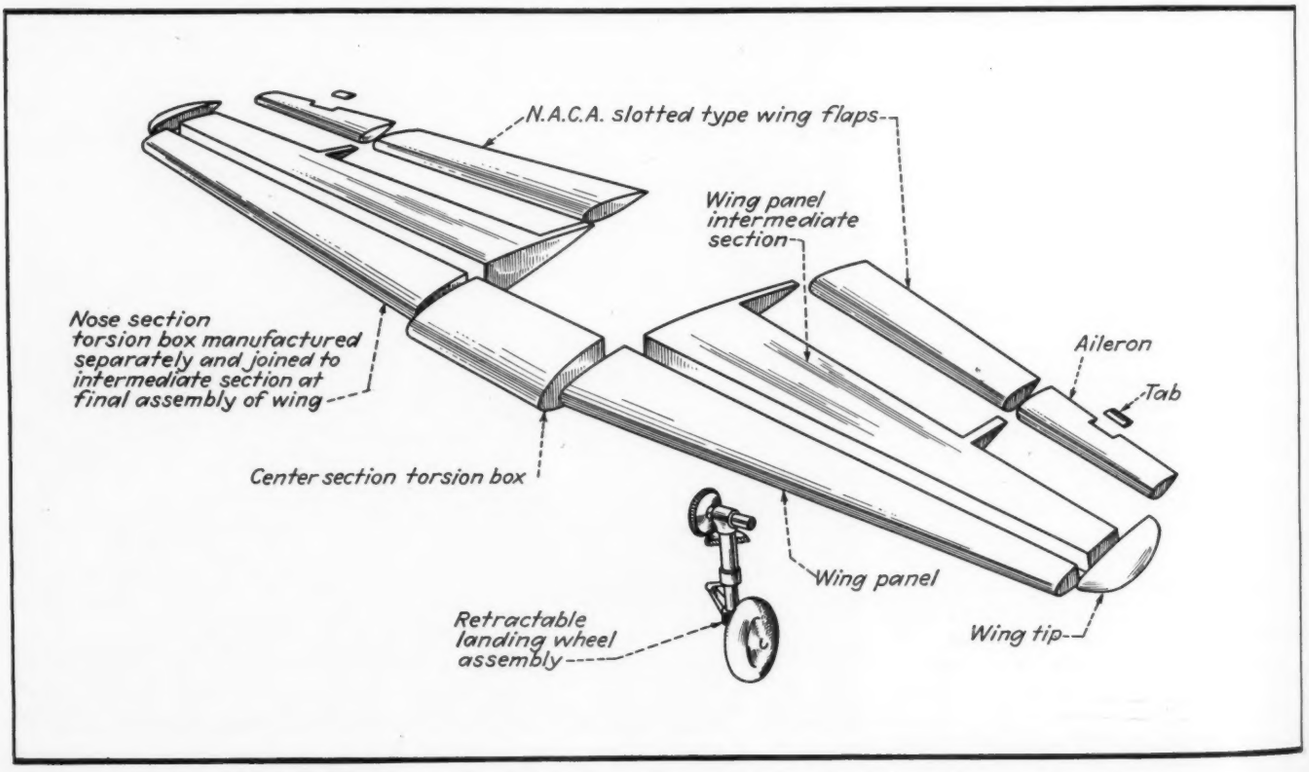
Production time can be saved if advance thought is given to the use of castings where practicable. Of course, the relative economy of sand, permanent mold, or die castings is determined by the quantity required and comparative costs. Generally the choice of a casting or of a forging for a given part is largely governed by the number of parts required, owing to the high cost of forging dies, while castings can be produced economically in limited quantities. Forgings may possess superior mechanical properties because of improved grain structure, but this is not always sufficient to justify the expense of forging when only a few pieces are required. Where the required mechanical properties make the use of a casting impractical and the quantity required does not justify a forging, the part should be machined from bar stock.

Weight Economy

Weight economy cannot be sacrificed in order to simplify production. The importance of saving even the minutest quantity of weight must not be underestimated, for cumulative savings on each of the several hundred assemblies making up the complete airplane structure will represent a sizeable total.

Engineers frequently specify unnecessarily heavy materials, particularly for parts of minor nature, such as nameplates, to which little attention is given in detail design. Such parts are frequently specified as aluminum alloy when magnesium, or possibly plastic would have ample strength, and weigh less. It should be remembered, however, that forming of magnesium alloy members frequently requires the use of expensive hot-forming equipment, and that production problems of magnesium alloys have to be investigated before deciding upon the use of this material. In many cases, it may be possible to use laminated phenolic sheet or fibre with wiped-in letters.

Analysis of each new design from the viewpoint of weight can best be accomplished by a careful examination of those factors known to affect weight economy. These are: substitution of aluminum or magnesium alloys for steel; substitution of plastics for metals, permitting up to 30 percent weight reduction; simplification of assembled parts; use of lighter sheet metal gages by judicious application of beads and stiffeners; increasing the size or quantity of lightening holes in sheet metal parts; removing all unnecessary mate-


N.A.C.A. slotted type wing flaps
Wing panel intermediate section
Nose section torsion box manufactured separately and joined to intermediate section at final assembly of wing
Aileron
Tab
Center section torsion box
Wing panel
Retractable landing wheel assembly
Wing tip

Monospar wing design for single-engine airplane, arranged for unit sub-assembly to speed production

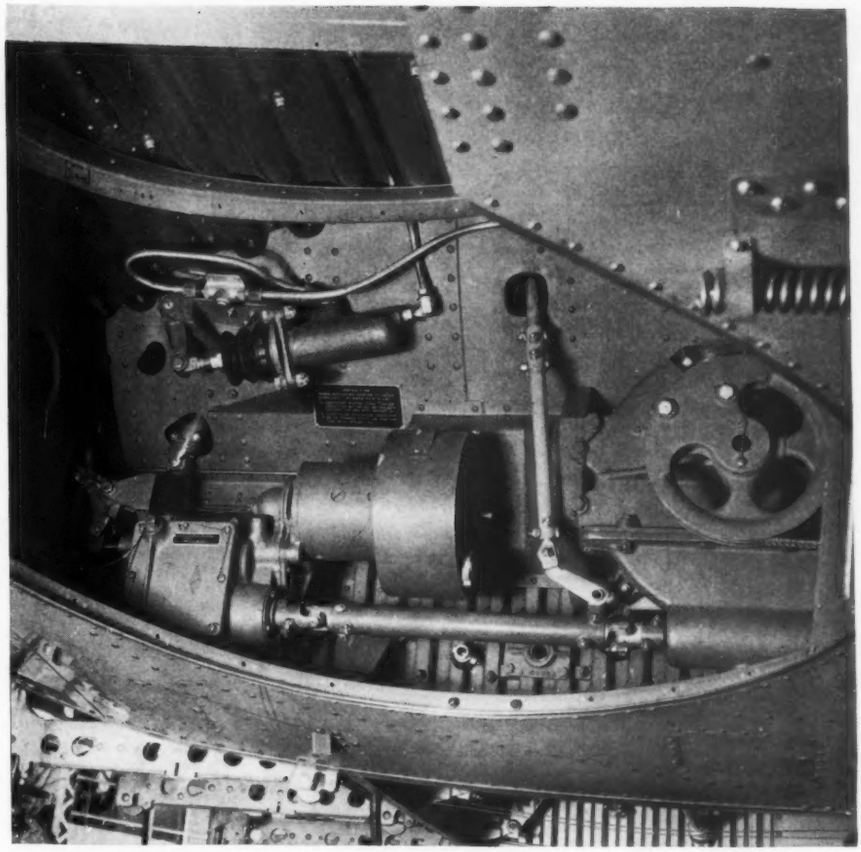
rial; substituting rolled or drawn sections for extruded sections; substituting forgings for castings; reducing the quantity of finish coats, or the thickness of platings.

Serviceability

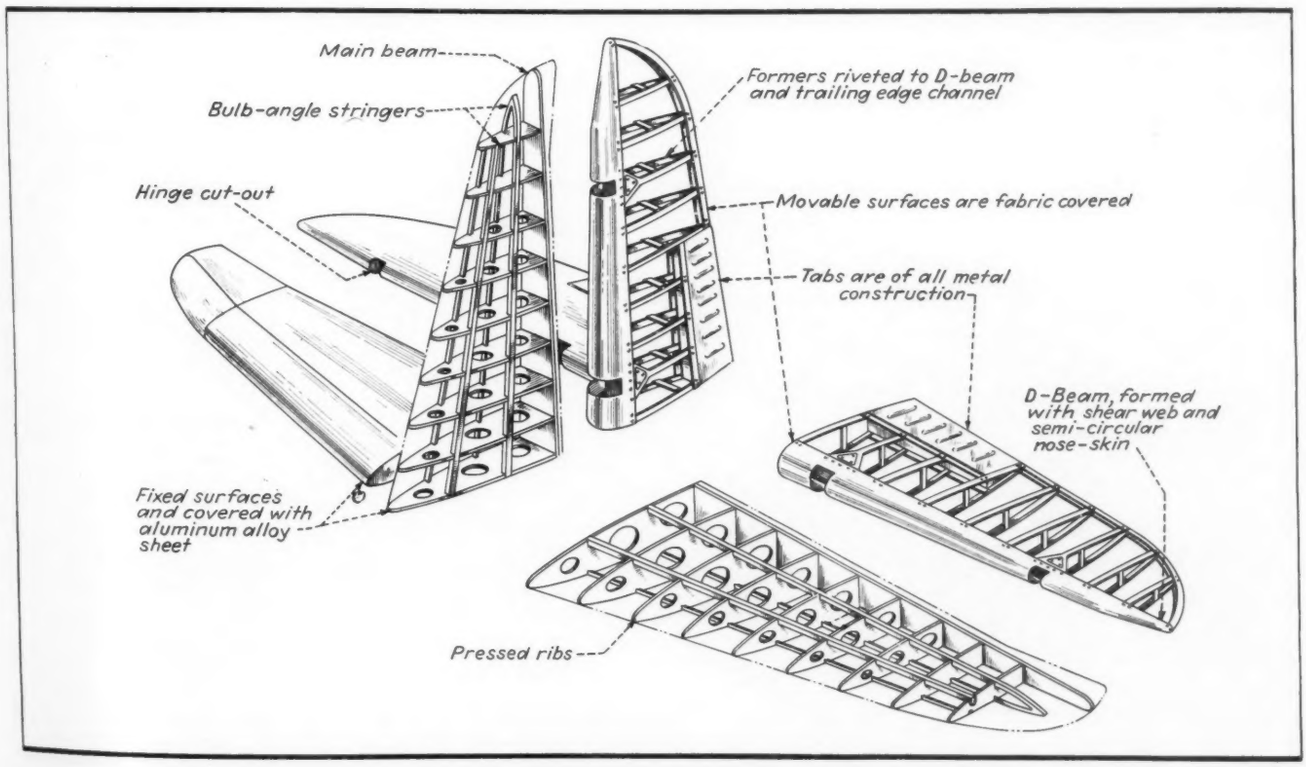
Throughout the development of every design, it must be remembered that a certain amount of inspection and maintenance will be required during the service life of the airplane, and the design should be such that these operations can be carried out easily and quickly. Serviceability is closely related to production economy as many of the governing factors are identical. It will be found, in general, that an airplane designed for economical production will be well adapted to service and maintenance operations.

The principal operations involved in inspection and maintenance are: inspection for wear or damage; lubrication; removal and replacement; repair or adjustment. Adequate means must be provided for complete, easy inspection of all mechanisms subject to wear, and for inspection of all structural elements which might be damaged by minor accidents to the airplane. Inspection doors should have quick-release fasteners rather than screws or bolts.

To reduce the number of points requiring periodic lubrication, self-lubricating bearings can be installed in all infrequently used or lightly loaded

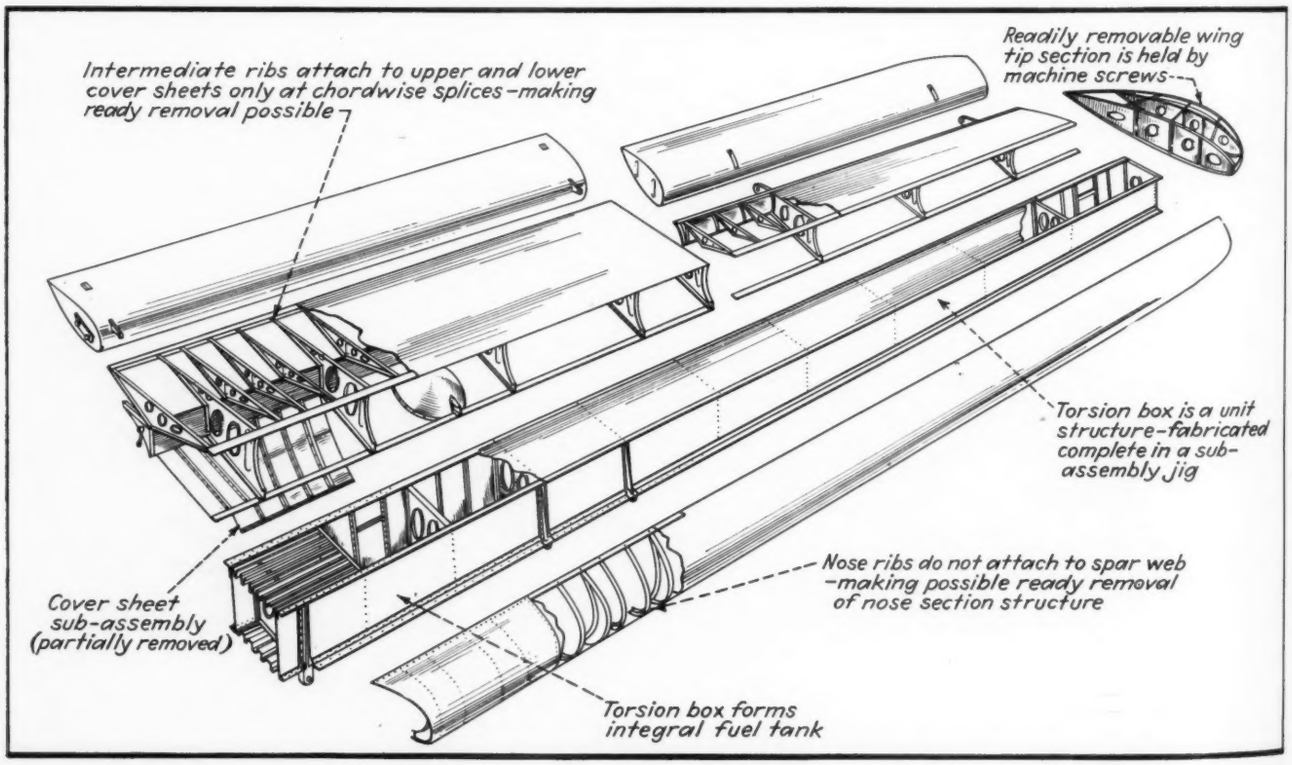


Accessibility for installation of sub-assemblies is important in obtaining production line economy. Complete access to landing gear and wing flap motor and transmissions of the Vultee V-12 Attack Bomber is obtained by placing these units within the landing gear left-side wheel-well


Main beam
Bulb-angle stringers
Hinge cut-out
Fixed surfaces and covered with aluminum alloy sheet
Formers riveted to D-beam and trailing edge channel
Movable surfaces are fabric covered
Tabs are of all metal construction
D-Beam, formed with shear web and semi-circular nose-skin
Pressed ribs

Simplified construction of empennage surfaces

Wing gun installation in Vultee V-12 Attack Bomber. Accessibility was the basic consideration in this design so as to insure quick installation, sighting and loading. Each gun compartment is secured to the wing with six readily removable capscrews


Intermediate ribs attach to upper and lower cover sheets only at chordwise splices–making ready removal possible
Readily removable wing tip section is held by machine screws
Torsion box is a unit structure–fabricated complete in a sub-assembly jig
Nose ribs do not attach to spar web –making possible ready removal of nose section structure
Cover sheet sub-assembly (partially removed)
Torsion box forms integral fuel tank

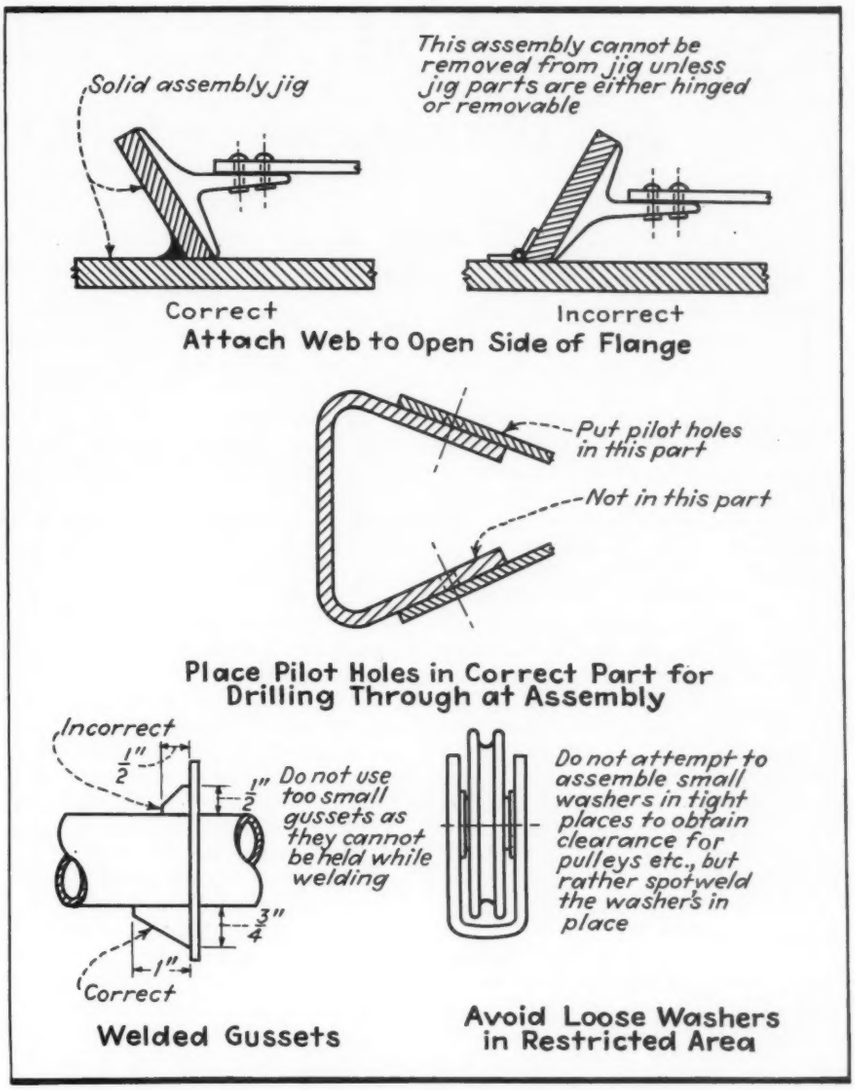
Construction of monospar wing for a multi-engine plane

bearings. Shielded, grease-packed, antifriction bearings are preferable where bearing loads are heavy. On the other hand, the designer should specify adequate lubrication for all friction surfaces where self-lubricating or antifriction bearings are not practicable. Adequate access for the application of the lubricant must be provided.

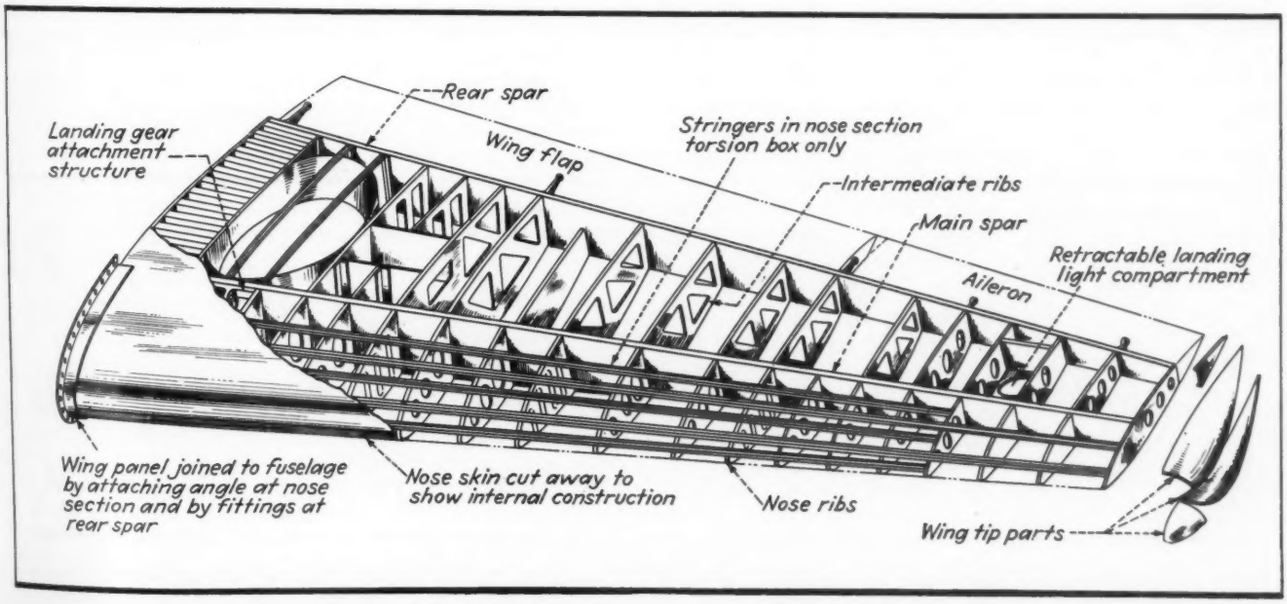
Usually the nature of the lubricant required is fairly obvious; in uncertain cases it is best to consult a lubrication engineer. The lubricant should be specified on the assembly or installation drawing as all places requiring a lubricant must be lubricated at final assembly of the airplane. Some discretion must be used, however, to avoid overlubrication. In a number of places such as cable fairleads lubrication will do more harm than good because the lubricant tends to collect grit and accelerate wear.

If proper consideration has been given to ease of assembly and adequate accessibility, little difficulty will be experienced with removal, replacement, or repair of parts. However, the fact that any part or unit of the airplane at any time may at some time have to be replaced or repaired must always be considered during the development of the design.

Valuable suggestions can be gained by consulting with shop supervisors on production practices, in order to arrive at the most economical construction methods from the consideration of production equipment and processes actually available in the factory. Be certain to give complete information about the matter in question to avoid possible misunderstandings.


Solid assembly jig
This assembly cannot be removed from jig unless jig parts are either hinged or removable
Correct
Incorrect
Attach Web to Open Side of Flange
Put pilot holes in this part
Not in this part
Place Pilot Holes in Correct Part for Drilling Through at Assembly
Incorrect
1/2"
1/2"
Do not use too small gussets as they cannot be held while welding
3/4"
1"
Correct
Welded Gussets
Do not attempt to assemble small washers in tight places to obtain clearance for pulleys etc., but rather spotweld the washers in place
Avoid Loose Washers in Restricted Area

Impractical assemblies which cause production delays and rejects


Rear spar
Landing gear attachment structure
Wing flap
Stringers in nose section torsion box only
Intermediate ribs
Main spar
Retractable landing light compartment
Aileron
Wing panel joined to fuselage by attaching angle at nose section and by fittings at rear spar
Nose skin cut away to show internal construction
Nose ribs
Wing tip parts

Construction of monospar wing of design illustrated on p. 282, for a single-engine plane, well adapted to fast production

BALL BEARINGS

For Thrust Loads in Centrifugal Pumps

THOMAS BARISH

Assistant Chief Engineer, Marlin Rockwell Corporation

THRUST LOADS in centrifugal pumps present important design problems which deserve thorough checkup both by theory and test. Two sources of thrust load need little attention; these are, the force necessary to change the flow of water from an axial to a radial direction, and the momentary starting shocks which occur when the column of water strikes the impeller. The latter is important only in some types of shallow deep well pumps.

By far the major source of thrust load is the difference in hydraulic pressure against the outer sides of the rotor; more specifically, the difference between the axial components of hydraulic thrust against the closed side of the rotor and that against the inlet side. Fig. 1 indicates how these pressures vary at different radial distances on an impeller with a single side suction opening. The net effect of the hydraulic pressure on the impeller is a large thrust load towards the suction side.

To illustrate the magnitude of these thrusts, test data are shown in Fig. 2 for a dredge pump in which the thrust load on the impeller was determined by measuring the unit pressures at various radial distances. ("Dredge Pump Pressures and Thrust Loads," James H. Polhemus and James Healy, A.S.M.E. *Transactions*, 1929, Vol. 51, HYD-51-4). With equal axial clearances of $\frac{3}{4}$ in. on both sides of a 70 in. dia. impeller, at 52 lb. per sq. in. discharge pressure, the thrust amounted to 30,100 lb. This thrust represents a 15 percent unbalance, that is, 15 percent of the outlet pressure acting against the total axial area of the closed side of the rotor.

The axial clearance determines the skin friction pumping effect on the sides of the impeller which is evident in the shape of the two curves in Fig. 2. Small axial shifts showed large changes in thrust load, caused by decreasing pressure on one side and increasing pressure on the other. Shifting the impeller axially in one direction brought the net thrust down to 15,000 lb., and shifting in the opposite direction increased the net thrust load to 36,000 lb. A similar dredge pump that was accidentally set up with insufficient clearance on the closed or back side reversed the direction of net thrust; the effect of which was clearly shown on the bearings when they were removed.

Note the relief holes shown in Fig. 2. These are important. Plugging these holes raised the thrust from 20,000 to 41,000 lb. These small holes, judiciously placed to reduce the thrust, permit cutting the size and the power losses of the bearing in half, thus more than justifying the resulting small increase in leakage losses. Tests made by Goulds Pumps, Incorporated, on small pumps indicated that for best results the total area of the relief holes should approximately equal the total clearance leakage area between the rotor and the case.

Few modern pump rotors have axial clearances for seals, chiefly because of thrust troubles resulting from axial shifts of the rotor. For sealing purposes most pumps have radial clearances like those shown in Fig. 3. Radial clearances can be made closer than axial clearances, they are more easily maintained, and permit larger running clearances between rotor and casing. Furthermore, the thrust load does not vary as greatly from pump to pump, nor does it change as much in any one pump because of wear or maladjustment.

With radial clearance seals the pressure distribution curves on the sides of the impeller become flatter and practically equal, also the pump thrust can be more nearly balanced. Numerous tests and service checks indicate that the maximum unbalance to be expected in such designs, including all other causes of thrust, is approximately 7 percent. This unbalance will drop to about 3 percent with careful design, with better maintenance of equal clearance on both sides of the rotor, and with liberal relief holes.

The diameters of the impeller at the seals, indicated by *A* and *B* in Fig. 3 are usually made identical for ease in machining and for interchangeability of clearance rings. However, *B* is sometimes made slightly larger than *A* in order to improve the balance, thus practically equalizing the thrust load. By increasing the diameter *B* the thrust can even be reversed, as shown by Fig. 4. Perfect balance, however, is difficult to maintain and usually unnecessary since

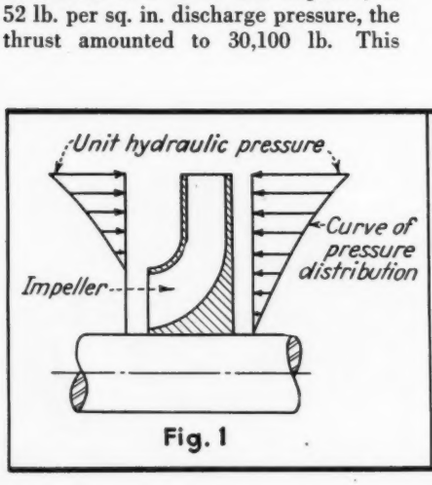


Fig. 1

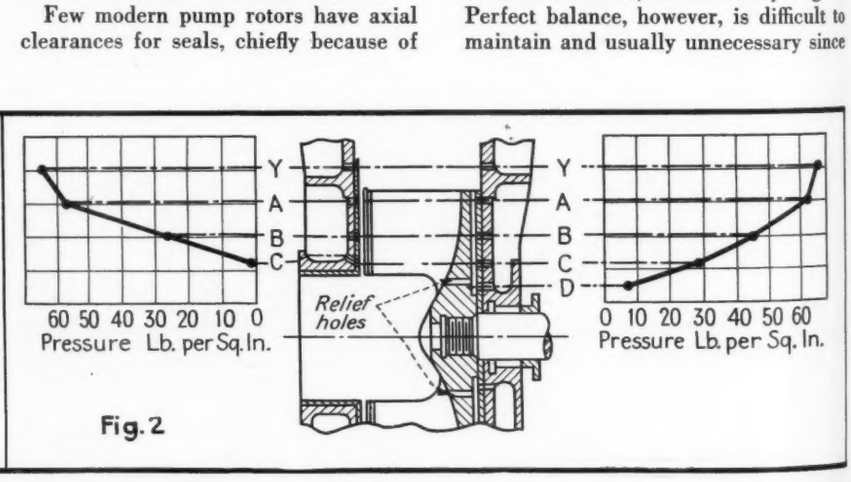


Fig. 2

Fig. 1—Curves of pressure distribution on the closed side and the inlet side of an impeller with single side suction, to illustrate how the total axial hydraulic pressure differs from side to side of the impeller thus causing a thrust toward the suction or inlet side. Fig. 2—Test data curves for a dredge pump, unit pressures are indicated for various radial distances

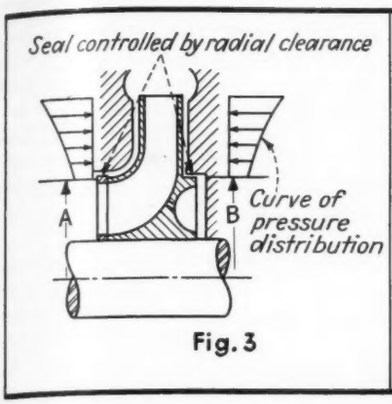


Fig. 3

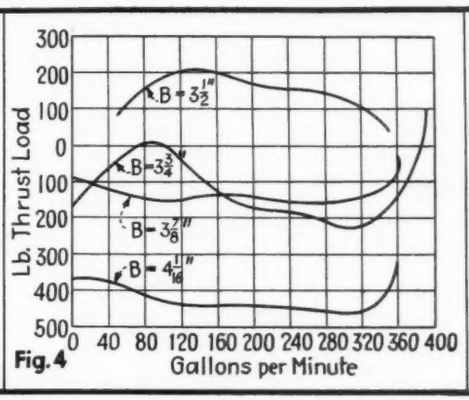


Fig. 4

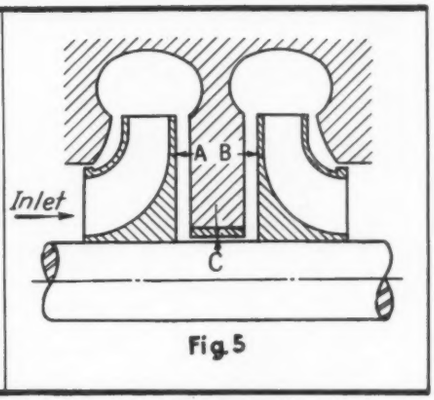


Fig. 5

Fig. 3—Impeller with radial seals, with this type of seal the distribution curves become flatter and practically equal. Fig. 4—Effect on thrust load caused by changing the diameter of the radial clearance at closed side of impeller. Fig. 5—Two-stage pump with impellers back to back is theoretically balanced, relief holes are not needed since pressures A and B are equal

a single row radial ball bearing will carry, without increase in size, the small residual unbalanced thrust in a well designed pump.

Multi-stage pumps introduce a new problem. A two stage pump, like that shown in Fig. 5, is theoretically balanced without resorting to clearance rings or relief holes since the total pressures in the directions of A and B should be exactly equal. However, absolute pressure at B is one stage higher than at A and any leakage through the shaft clearance C will lower B and increase A, thus tending to build up the thrust. As the pump wears, clearance C increases and the thrust gradually becomes larger with age. Pumps of this type should be checked regularly for worn clearance rings. Too long a wait may make it necessary to replace the thrust bearings as well as the rings. Some manufacturers make the diameter of the high-pressure rotor slightly larger than that of the low-pressure rotor, thereby reducing or reversing the initial thrust, thus permitting far more wear at the clearance rings before the thrust load mounts dangerously.

Deep well pump design must contend with another problem that involves thrust; first, to the thrust load of the impeller must be added the shaft and rotor weights; second, the long shaft elongates appreciably as the pump starts when the hydraulic thrust develops. Frequently the rotor drops as much as ¼ in. so that the pump design is complicated by the necessity of providing long axial clearances. Third, deep well pumps frequently handle water containing abrasive such as fine sand. These last two difficulties make clearance rings and relief holes impractical. Hence thrust loads become quite large.

Fortunately, however, such thrusts can be accurately calculated. The axial clearances are so large that pressure curves like those of Fig. 1 are flat and at maximum value.

With H = head in ft. (total, all stages)
D = dia. of impeller inlet in in.
d = shaft dia. in in.

The thrust load is

$$T = \text{weight of rotating parts} + 0.34\,H\,(D^2 - d^2)$$

For screw type impellers, D becomes the diameter of the impeller.

Designers differ widely in their opinions concerning the importance of making provisions to support radial loads. Experience indicates that radial loads are usually negligible. Theoretically they consist of:

(1) Weight of parts
(2) Unbalanced mechanical loads
(3) Momentary shocks when handling a percentage of solids
(4) Hydraulic unbalance or more exactly reactions from inlet and exhaust scroll conditions or starting surges.

None of these are appreciable in most centrifugal pumps except occasionally unbalanced loads caused by poor machining or mounting, but these need not be tolerated. Before bearing difficulties arise conditions of unbalance are quickly indicated by the occurrence of noise or vibration, and packing trouble caused by shaft deflections.

In certain services bad erosion is unavoidable; then expected large mechanically unbalanced loads necessitate more radial capacity and also greater shaft stiffness.

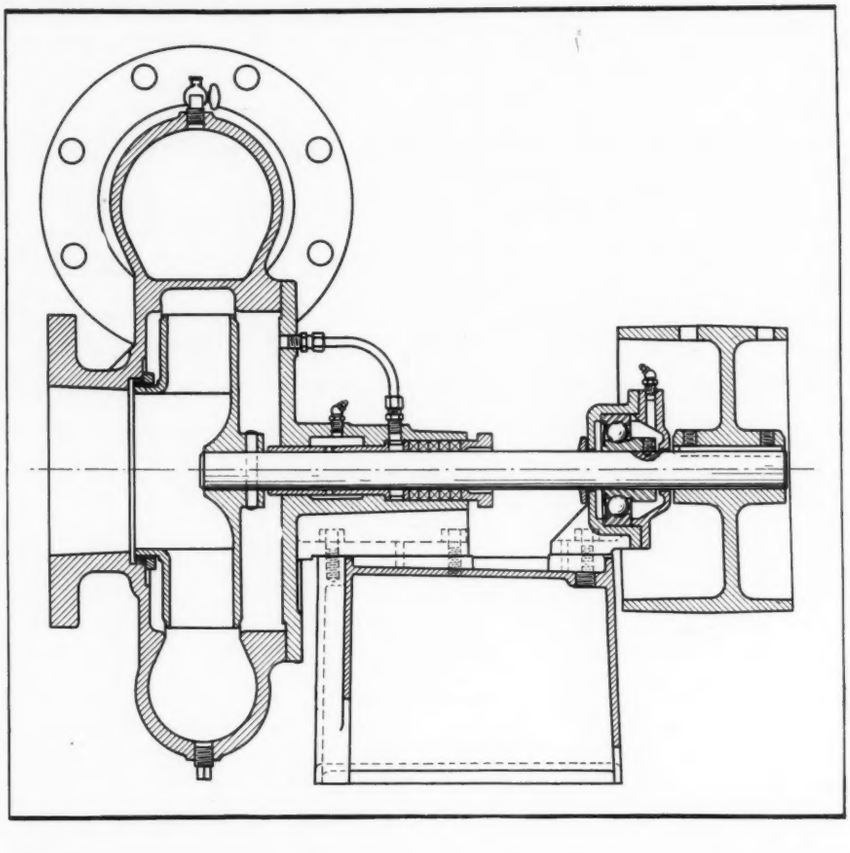
Shaft design controls the bearing size more than the loads do, this practice provides the most definite proof that radial loads are not a direct design consideration. The pump design itself requires a compromise in shaft size, since for better inlet conditions at the rotor, and for lower rubbing speeds at the packing gland the shaft should be small; but for stiffness and critical speeds even with relatively long centers the shaft should be large. Once this compromise is reached, a single-row light series ball bearing of a size to fit the shaft usually provides more than enough capacity.

Shafts for deep well pumps are long as compared with their diameters so that any radial load approaching the capacity of a single-row light series ball bearing would seriously overstress the shaft in bending at the point of loading. Where the loads are considerable, large shaft diameters are necessary, and a light series single-row bearing of correspondingly increased bore is ample. This point is emphasized because of the great divergence of opinion between pump manufacturers concerning radial bearings. Many pumps are operating satisfactorily that use only light series single row ball bearings. These pumps differ but little from designs using larger bearings and even double row bearings. One important exception must be noted, namely where bearing capacity must be provided for belt loads, especially when overhung.

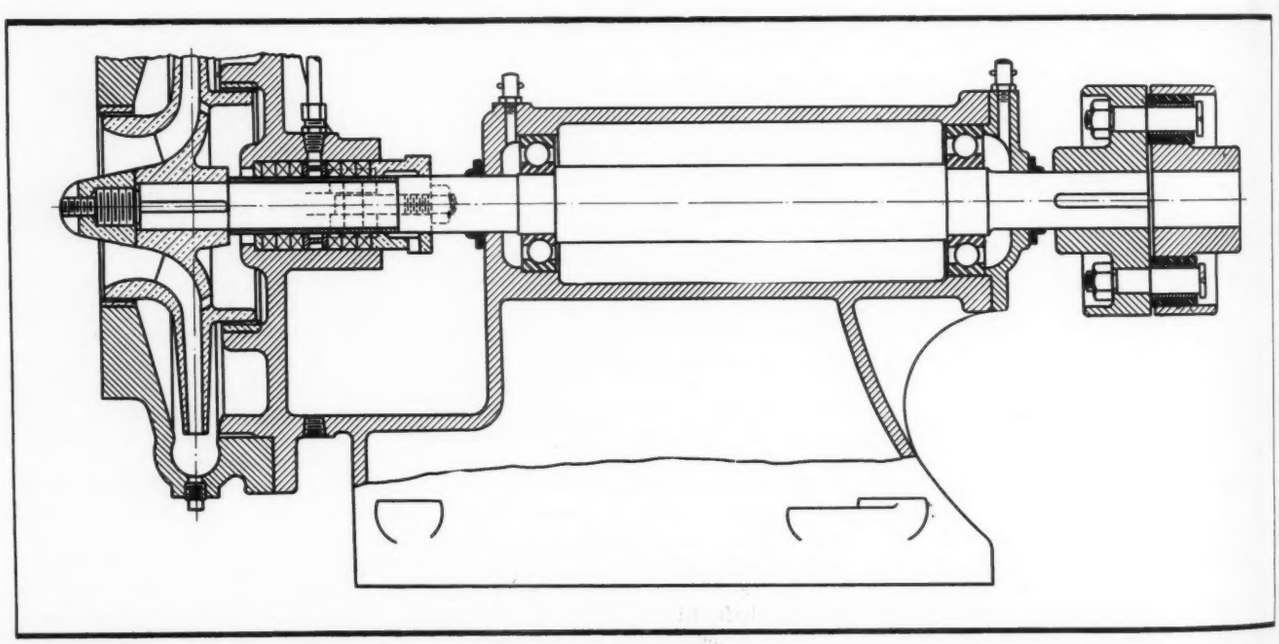
Shock loads, particularly starting surges, probably are absorbed by the packing glands because of shaft flexibility. The portion of the load that does reach the ball or roller bearings rarely needs checking, because bearings which are selected for fatigue life under normal high-speed conditions have a large capacity for short duration shocks.

The typical centrifugal pump designs presented on the next two pages are included to illustrate some of the bearing problems encountered. Ball and roller bearings are particularly advantageous in centrifugal pumps not only because of the small running clearances involved, but also because of the negligible wear under normal operating conditions and their relatively low sensitivity to shaft deflection and also to housing tilt.

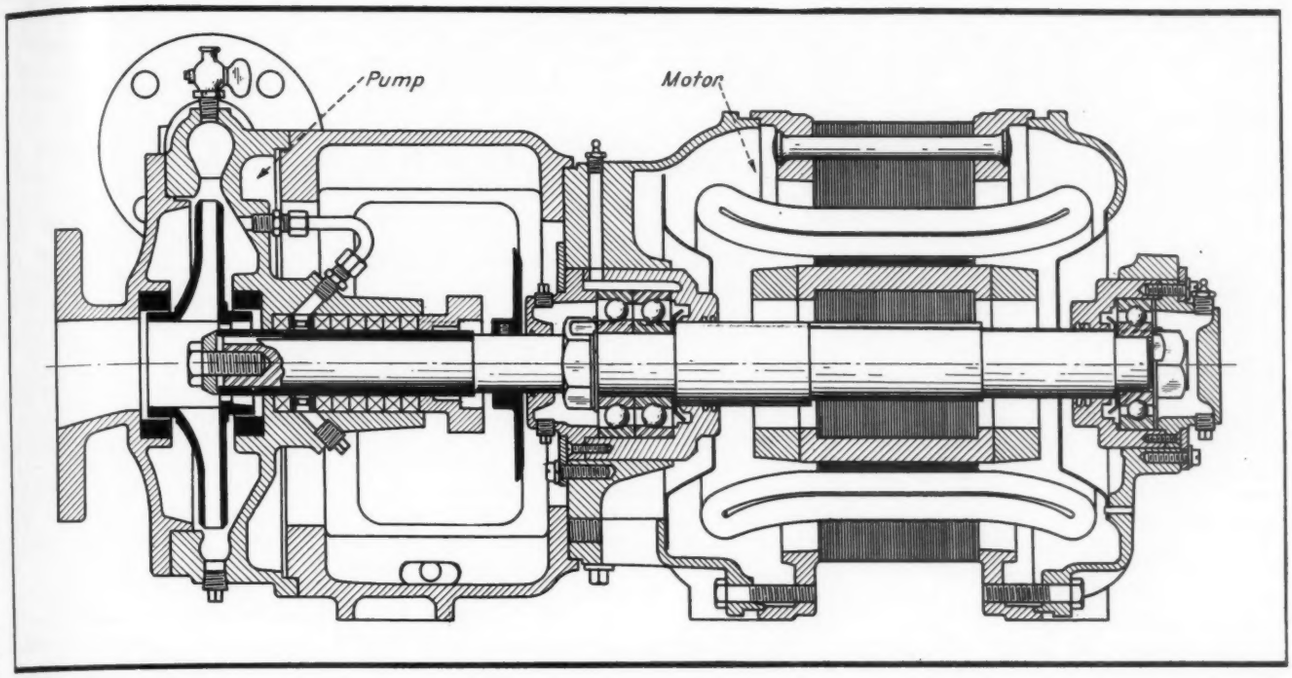
(Continued on next page)



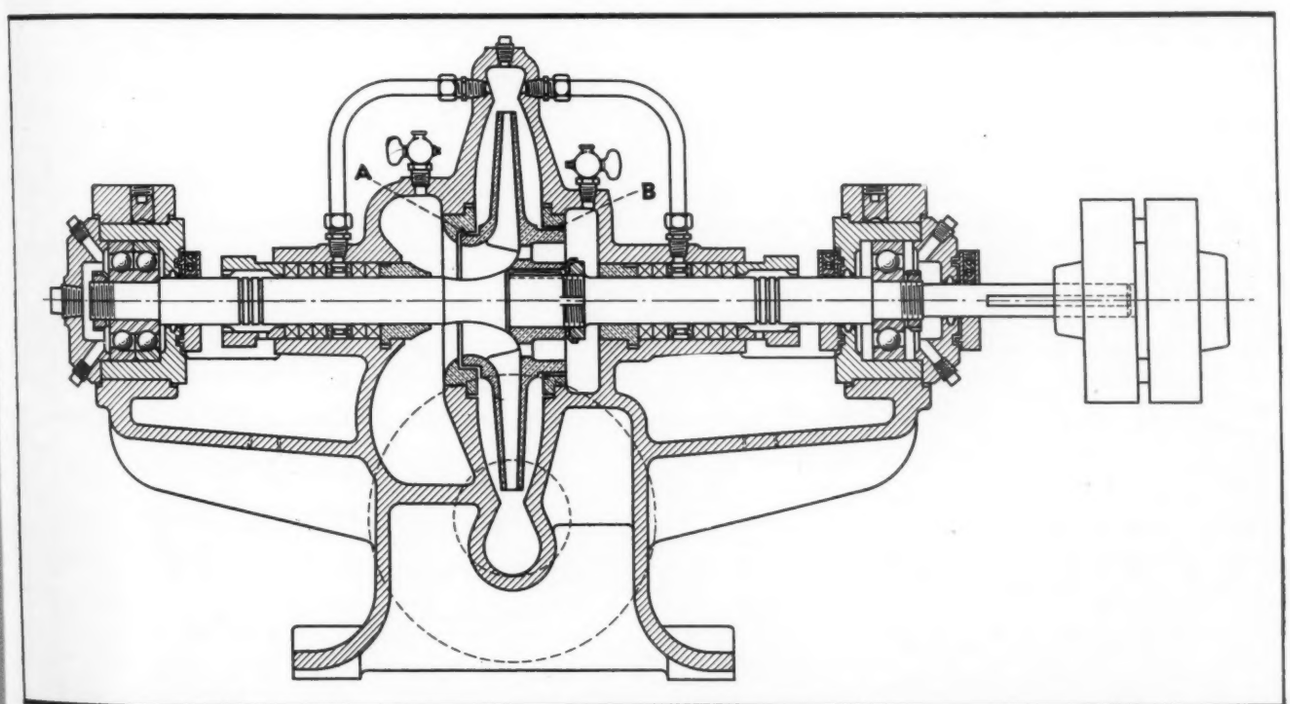
Typical low cost pump designed for intermittent service, uses a water lubricated bushing for the rotor bearing. At the pulley end of shaft is a special ball bearing with wide inner ring which permits the use of a straight shaft with no shoulders and practically no machining. The extremely close limits of modern centerless stock, as low as 0.0005 in. tolerance, is combined with close bearing bore tolerances (+ 0.0000 — 0.0005 in.) to give theoretical fits from 0.0003 tight to 0.0007 in. loose. These fits with wide bearings are acceptable for intermittent service, especially since the maximum tightness or looseness of fit rarely occurs. The inner ring of the bearing is fastened by set screws which can be reached through the oiling hole or by removing the cap. Note how the bearing is placed well under the pulley to reduce overhang, thus keeping the load small on the bushing. The bearing is amply protected against leakage by an integral shield, a liberal auxiliary "trap" space, a close shaft clearance in the housing, and an external rubber slinger. A popular alternate design uses a standard bearing pressed on a machined sleeve, fastened to the shaft by set-screws.



Overhanging the rotor eliminates one packing gland, simplifies the piping to the inlet and reduces machining. The narrowness of ball bearings permits designs in which rotors can be overhung in single stage pumps and many two stage pumps. In the design shown the bearings are fitted in a one piece housing to assure rigidity and initial alignment. The larger O.D. of the single-row thrust bearing permits performing all machining operations at one setting. Earlier designs used separate bearing housings but the single housing eliminated two seals and simplified machining. The larger lubricant space is not a disadvantage. For two-stage pumps or other types that may have larger thrust loads, Duplex or double-row ball bearings can be used in the place at which endwise movement is to be resisted. Large dredging pumps are also of this over-hung type mounted on roller bearings next to the rotor. The thrust bearing may be a Kingsbury, or a Duplex bearing carrying radial and thrust load. One design uses a 1½ pair Duplex ball bearings at the drive end with two angular contact type bearings in parallel which divide the main thrust load, and the third bearing to take occasional reverse thrust and to keep the other pair tight.



Advanced designs in centrifugal pumps are "close coupled," that is, mounted directly on the motor shaft. This compact combining of the two high-speed elements, pump and motor, eliminates two bearings, a coupling, and belts. The type has proved successful. The motor manufacturer faces a few difficult problems, such as keeping the long shaft extension true running in spite of locknuts tending to spring the shaft, maintaining squareness at the face where the pump casting is bolted, controlling axial locating dimensions, and also providing bearing capacity for the pump. Bearing requirements do not vary from those for other pump types described as the motor adds only a small radial load. The Duplex, or double-row angular contact type, ball bearing provides the large thrust capacity needed for two stage pumps, since the same motor is used to drive one and two stage pumps. This thrust carrying bearing can be at either end of the motor. The motor manufacturer also maintains for this service the close control of bearing fitting, alignment and lubrication found necessary for other high-speed motor applications.



Straddle mounted continuous duty centrifugal pump with single-row ball bearings at the drive end and an angular contact type double row at the closed end to provide large thrust capacity. With the refinements in rotor design for minimum thrust load, a single-row ball bearing would be sufficient at the thrust end. Interchangeable "cartridge" type bearing housings permit removal of rotor and shaft assembly without exposing bearings.

BENDING STRESSES

In Materials Having Dissimilar Elastic Moduli

WILLIAM I. POWELL, *Assistant Mechanical Engineer*
Board of Water Supply of the City of New York

CLOSELY related to the subject of stress distribution in clad materials, which was discussed in the article "Composite Members, How to Find the Stress in Each Material," presented in the May number of PRODUCT ENGINEERING, is the analyses of stress distribution in materials having dissimilar moduli in tension and in compression.

The analysis that will be set forth in this article was made by the author to justify the results of a series of tests on manganese bronze in which surprising variations were found to exist in the physical characteristics of the metal, the most interesting of which was the existence of dissimilar moduli.

The tension curve for one of the bars pulled is shown in Fig. 1. Inspection of this curve shows that the material falls into the category of non-elastic metals. Because of this fact all working stresses must be retained well within the proportional line. An arbitrary conservative value for the working stress in tension was chosen as 5,000 lb. per sq. in. This was also necessitated by the tendency of the metal to creep, even at comparatively low temperatures.

A section of the same bar was used to obtain the compression curve shown in Fig. 2. The practically straight nature of this graph indicates a much higher ultimate strength in compression than that in tension. For one specimen the ultimate compression strength was found to be twice that in tension. It will also be noticed on comparing Figs. 1 and 2 that E_c, the modulus of elasticity in compression, is about 1.4 E_t, the modulus of elasticity in tension, indicating a non-isotropic material.

Another section of the same bar was tested for deflection as a simple beam loaded at the center as shown in Fig. 3. The graph in Fig. 4 shows the results of this transverse bend test.

The ratio of the unit stresses, as calculated in Fig. 4 is less than the proportional ratio of the two moduli, thus departing from that which would normally obtain in a material with a single modulus. Although this condition is encountered rather infrequently, when it does occur it involves a problem in flexure of considerable difficulty to machine designers. Therefore, chiefly because of the paucity of detailed information on the subject, it is the author's intention to set forth clearly, through an original approach, a picture of what he believes to be the effect of bending loads on materials having dissimilar moduli.

The effect on the strength properties of such a metal, the inelasticity of which always presents a problem in the judicious selection of a unit stress, may be far-reaching. In calculations for plates or pressure vessels involving two dimensional stress, the modulus of elasticity E occurs frequently and possibly has a cumulative effect on the value of the true stresses. The apparently inconsistent differences found may be the result of a number of causes, some of which are as yet quite obscure. The unknown factors undoubtedly involve the degree of chemical homogeneity produced by casting conditions such as pouring temperature, allowable time in the molds and possible intermolecular effect of unequal cooling in bulky or thick sections.

In the calculation of bending stresses in any material having a single elastic modulus in tension and compression, the usual stress formula $S = Mc/I$ will give the correct value for the extreme fibre stress. This, however, is found not to be true when two separate moduli for tension and compression exist. In the specimens tested the disparity between

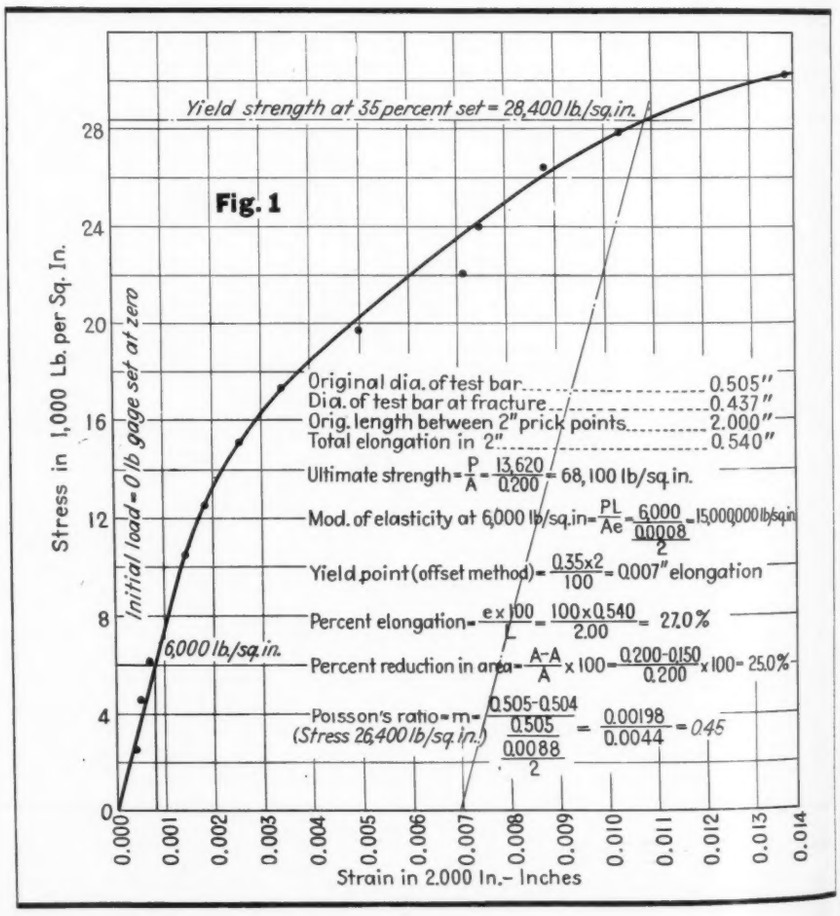


Fig. 1
Yield strength at 35 percent set = 28,400 lb./sq.in.
Stress in 1,000 Lb. per Sq. In.
Initial load = 0 lb gage set at zero
6,000 lb./sq.in.
Original dia. of test bar ..... 0.505"
Dia. of test bar at fracture ..... 0.437"
Orig. length between 2" prick points ..... 2.000"
Total elongation in 2" ..... 0.540"
Ultimate strength = P/A = 13,620 / 0.200 = 68,100 lb/sq.in.
Mod. of elasticity at 6,000 lb/sq.in = PL/Ae = 6,000 / (0.0008/2) = 15,000,000 lb/sq.in
Yield point (offset method) = 0.35x2/100 = 0.007" elongation
Percent elongation = e x 100/L = 100 x 0.540/2.00 = 27.0%
Percent reduction in area = (A-A)/A x 100 = (0.200-0.150)/0.200 x 100 = 25.0%
Poisson's ratio = m = (0.505-0.504)/0.505 / (0.00088/2) = 0.00198/0.0044 = .045
(Stress 26,400 lb/sq.in.)
0 4 8 12 16 20 24 28
0.000 0.001 0.002 0.003 0.004 0.005 0.006 0.007 0.008 0.009 0.010 0.011 0.012 0.013 0.014
Strain in 2.000 In.- Inches

Fig. 1—Tension test curve of manganese bronze specimen

the two moduli was found to vary considerably. Figs. 1 and 2 show the maximum variation.

The tension test curve shown in Fig. 1 yielded a value of 15,000,000 for E_t, while a 5 in. compression section cut from one end of this bar prior to machining. gave an E_c of 21,000,000, see Fig. 2. From this bar, originally 14 in. in length. a piece was used for the bend test plotted in Fig. 4. The tension and compression specimens were cut from unstrained ends of the original bar.

From the load deflection curve Fig. 4. it was found that for a load of 500 lbs. the resulting deflection was 0.0019 in. at the center of the span. This load and deflection upon insertion in the deflection formula

$$f = \frac{W\ L^3}{EI\ 48}$$

taking the E_t as 15,000,000 was found to produce an effective moment of inertia I for the section which was less than the actual I of the rectangular section or $b\ d^3/12$. This decrease in I from $b\ d^3/12$ indicated a shift of the neutral axis to the side offering the greater resistance to load, or the compression side in this case as indicated in the tests.

The beam was then considered to be composed of two materials of dissimilar elasticity, one in tension and the other in compression, with the line of cleavage at the neutral axis. If this is assumed to be true the actual areas in tension and compression will bear a certain relationship to the effective areas under those stresses. This relationship is in direct proportion to the ratio of the two moduli, or assuming the larger compression modulus as unity, the smaller tension modulus will act on an effective area equal to $(E_t/E_c) \times$ *the actual tension area.* For simplicity's sake the effective area figure chosen is that of a trapezoid. This shape lent greater ease and facility to the resulting derivations

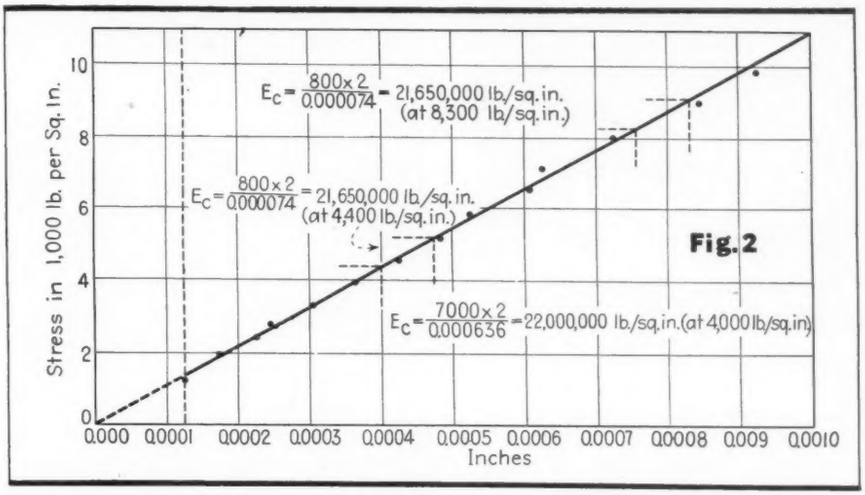

Stress in 1,000 lb. per Sq. In.
Inches
$E_c = \frac{800 \times 2}{0.000074}$ = 21,650,000 lb./sq. in. (at 8,300 lb./sq. in.)
$E_c = \frac{800 \times 2}{0.000074}$ = 21,650,000 lb./sq. in. (at 4,400 lb./sq. in.)
$E_c = \frac{7000 \times 2}{0.000636}$ = 22,000,000 lb./sq. in. (at 4,000 lb./sq. in.)
Fig. 2

Fig. 2—Compression test curve of manganese bronze specimen

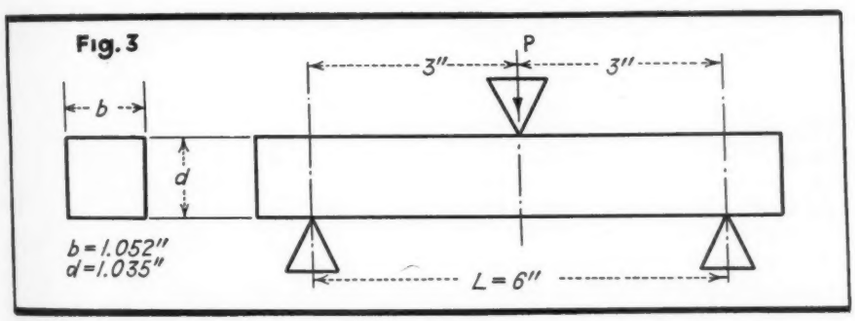


Fig. 3—Method of loading manganese bronze bar for transverse bending test

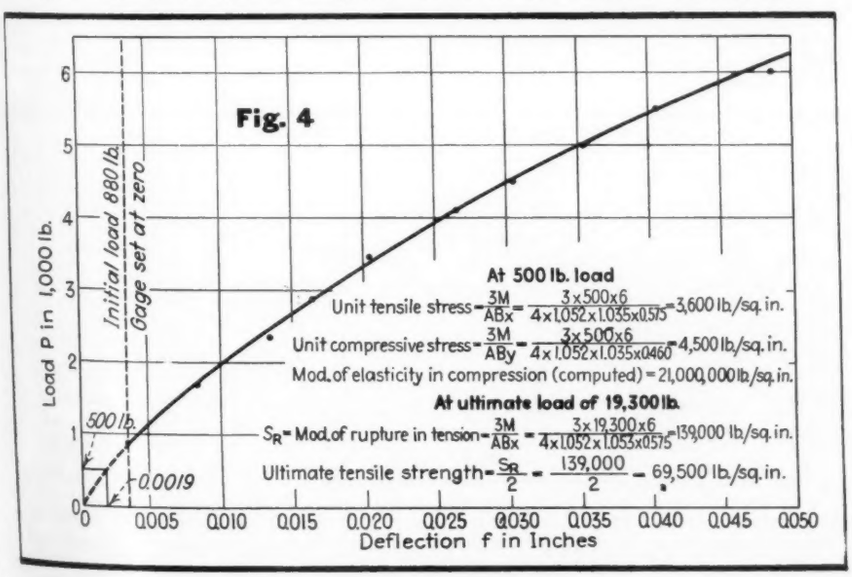


Fig. 4
Load P in 1,000 lb.
Deflection f in Inches
Initial load 880 lb.
Gage set at zero
500 lb.
0.0019
At 500 lb. load
Unit tensile stress = $\frac{3M}{ABx} = \frac{3 \times 500 \times 6}{4 \times 1.052 \times 1.035 \times 0.515}$ = 3,600 lb./sq. in.
Unit compressive stress = $\frac{3M}{ABy} = \frac{3 \times 500 \times 6}{4 \times 1.052 \times 1.035 \times 0.460}$ = 4,500 lb./sq. in.
Mod. of elasticity in compression (computed) = 21,000,000 lb./sq. in.
At ultimate load of 19,300 lb.
S_R = Mod. of rupture in tension = $\frac{3M}{ABx} = \frac{3 \times 19,300 \times 6}{4 \times 1.052 \times 1.053 \times 0.515}$ = 139,000 lb./sq. in.
Ultimate tensile strength = $\frac{S_R}{2} = \frac{139,000}{2}$ = 69,500 lb./sq. in.

Fig. 4—Curve of transverse bend test

and in the end yielded surprisingly accurate results, as a check with the methods of other authors revealed.

To develop formulas which would help in calculating the bending stresses in beams made of materials which have dissimilar elastic moduli, a group of symbols were adopted in which,

E_c = Modulus of elasticity in compression
E_t = Modulus of elasticity in tension
A_c = Actual area in compression
A_t = Actual area in tension
a_c = Effective area in compression
a_t = Effective area in tension
a_e = Total effective area
M_a = External bending moment
M_e = Effective resisting moment
T = Tensile forces on area A_t
C = Compressive forces on area A_c
S_t = Maximum unit tensile stress
S_c = Maximum unit compressive stress

Using these symbols, formulas are here developed for two conditions: *(a)* when the modulus in compression is greater than that in tension, and *(b)* when the modulus in tension is greater than that in compression. For both conditions it is assumed that the true beam is replaced by one having the same modulus throughout and an effective area of trapezoidal shape having a depth equal to the depth of rectangular section of the true beam.

CASE 1. Modulus of elasticity in compression is greater than modulus of elasticity in tension.

For this condition when under load the material yields more on the tension

side than on the compression side, therefore, as shown in Fig. 5, the trapezoid of effective area has a neutral axis which is above the neutral axis of the true beam section; in which

$a_e = a_c + a_t$, and $A_e = b\,y$

by construction

$$a_e = A_c = b\,y = y\left(\frac{b_1 + b_3}{2}\right)$$

from which

$$b_1 + b_3 = 2\,b \qquad (1)$$

$$b_3 = b_1 - \frac{y}{d}\,(b_1 - b_2) \qquad (1a)$$

also

$A_t = bx$, and

$$a_t = A_t\,(E_t/E_c) = bx\,(E_t/E_c) = \frac{x\,(b_2 + b_3)}{2}$$

therefore

$$b_2 + b_3 = 2b\,(E_t/E_c) \qquad (2)$$

combining Equations (1) and (2)

$$b_1 - b_2 = 2b\,(1 - E_t/E_c) \qquad (3)$$

Further

$$a_e = a_c + a_t = b\,(y + E_t x/E_c) = (b_1 + b_2)\,d/2$$

from which

$$b_1 + b_2 = (2\,b/d)\,(y + E_t x/E_c) \qquad (4)$$

combining Equations (3) and (4)

$$y + E_t x/E_c = \frac{d}{b}\left[b_1 - b\left(1 - \frac{E_t}{E_c}\right)\right] \qquad (5)$$

Substituting for y in Equation (5) the value of $(d–x)$

$$b_1 = \frac{b}{d}\left[x\left(\frac{E_t}{E_c} - 1\right) + d\right] + b\left(1 - \frac{E_t}{E_c}\right) \qquad (6)$$

$$\text{or } E_c = \frac{bE_t}{d}\left(\frac{x - d}{b_1 - 2b + bx/d}\right) \qquad (7)$$

$$\text{and } E_t = \frac{d\,E_c}{b}\left(\frac{b_1 - 2b + bx/d}{x - d}\right) \qquad (8)$$

From the geometry of the assumed figure

$$x = \frac{d\,(b_2 + 2b_1)}{3\,(b_1 + b_2)}, \text{ or } b_2 = \frac{b_1\,(2d - 3x)}{3x - d} \qquad (9)$$

also $T = C = S_t\,bx/2 = S_c by/2$

$$M_a = M_c = \frac{2}{3}\,(C \text{ or } T)\,(y + x) \qquad (10)$$

$$S_c = \frac{3\,M_a}{b\,(y^2 + xy)} = \frac{3\,M_a}{bdy} \qquad (11)$$

and since $S_t = S_c(y/x)$

$$S_t = \frac{3\,M_a}{bdy}\,\frac{y}{x} = \frac{3\,M_a}{bdx} \qquad (12)$$

CASE II. Modulus of elasticity in tension is greater than modulus of elasticity in compression.

For this condition when under load the material yields more on the compression side than on the tension side, therefore, as shown in Fig. 6, the trapezoid of effective area has a neutral axis which is below the neutral axis of the true beam section.

$$b_1 = \frac{b}{d}\left[x\left(\frac{E_c}{E_t} - 1\right) + d\right] + b\left(1 - \frac{E_c}{E_t}\right) \qquad (13)$$

$$E_t = \frac{bE_c}{d}\left(\frac{x - d}{b_1 - 2b + bx/d}\right) \qquad (14)$$

$$E_c = \frac{dE_t}{b}\left(\frac{b_1 - 2b + bx/d}{x - d}\right) \qquad (15)$$

$$S_c = \frac{3\,M_a}{bdx}, \text{ (16); and } S_t = \frac{3\,M_a}{bdy} \qquad (17)$$

The verification of E_c with the results obtained in the tests is a secondary use of the formulas. Where the two moduli have been predetermined, the sequence is as follows: trial values of x must be inserted in Equations (6) or (13) as the case may be, to obtain b_1, after which b_2 may be found from Equation (9), b_3 may now be found by Equation (1a), after which Equation (1) is used as a check on the value of $2b$. When both sides of Equation (1) balance, the correct value of x has been chosen.

PROBLEM. To check $E_c = 21{,}000{,}000$ as produced by the compression test by means of the transverse bend test data. Load $P = 500$ lb. and the corresponding deflection $f = 0.0019$ in. E_t is given as 15,000,000. In Fig. 7 is shown the true section of the beam and also the assumed effective area.

$$\text{Actual } I_g \text{ of section} = \frac{1.052 \times 1.035^3}{12} = 0.0973 \text{ in.}^4$$

Max. unit stress assuming no shift in neutral axis

$$S_{max.} = \frac{3PL}{2\,bd^2} = \frac{3 \times 500 \times 6}{2 \times 1.052 \times 1.035^2} = 4{,}000 \text{ lb. per sq. in.}$$

Taking $E_t = 15{,}000{,}000$

$$\text{Effective } I_e \text{ of section} = \frac{PL^3}{48\,E_t f} = \frac{500 \times 6^3}{48 \times 15{,}000{,}000 \times 0.0019} = 0.0788$$

$I_e = I_g$ of the trapezoid, or

$$I_e = \frac{d^3\,(b_1^2 + 4b_1b_2 + b_2^2)}{36\,(b_1 + b_2)}$$

For a trial value let $b_1 = 1.185$ in., then

$$0.0788 = \frac{1.035^3\,(1.185^2 + 4 \times 1.185 b_2 + b_2^2)}{36\,(1.185 + b_2)}$$

from which $b_2 = 0.588$ in.

Putting this value of b_2 in Equation (9)

$x = 0.575$ in.

$y = d - x = 0.460$ in.

From Equation (1a), $b_3 = 0.920$ in.

$b_1 + b_3 = 1.185 + 0.920 = 2.105$ in.

$2b = 2 \times 1.052 = 2.104$ in.

which is sufficiently close.

From the tests E_c is greater than E_t, therefore, from Equations (7), (11) and (12)

$E_c = 21{,}000{,}000$

$S_t = 3{,}600$ lb. per sq. in.

$S_c = 4{,}500$ lb. per sq. in.

The formulas derived herein are applicable to rectangular sections in bending, and are suited to either single or two dimensional stress solutions.

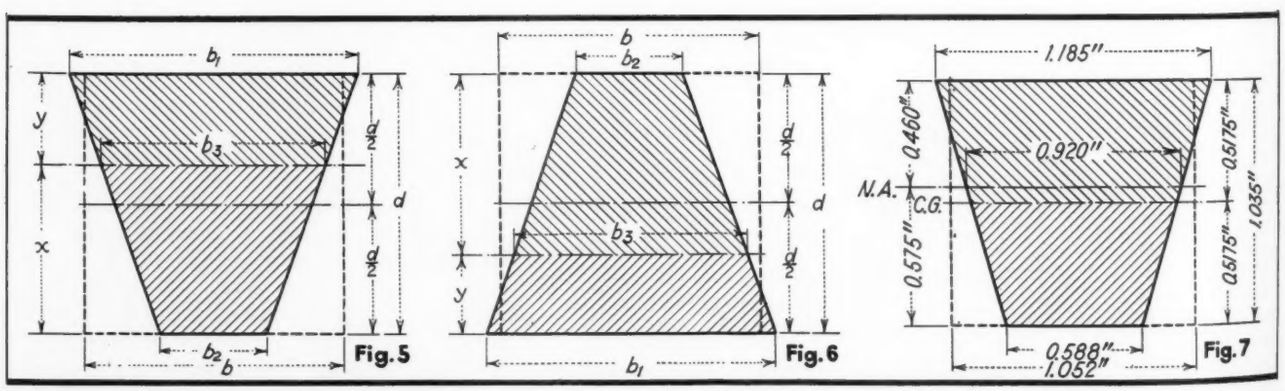
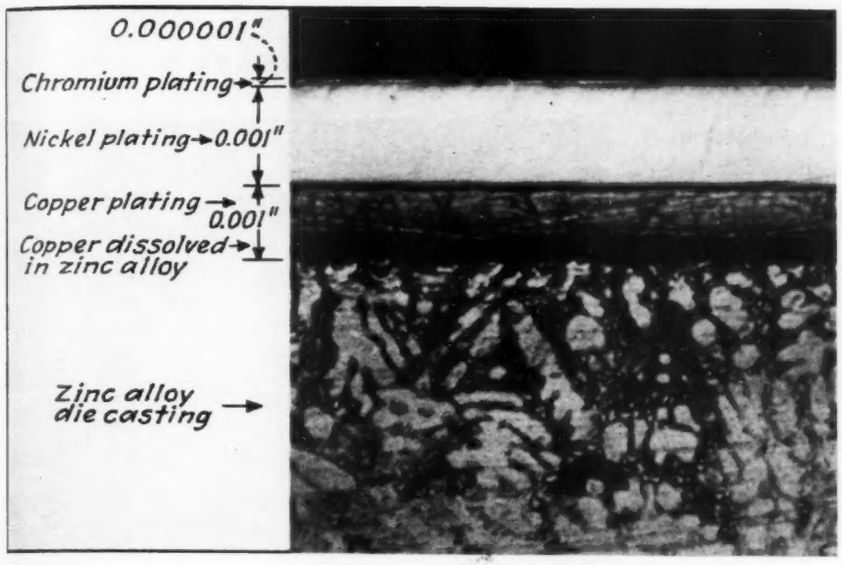
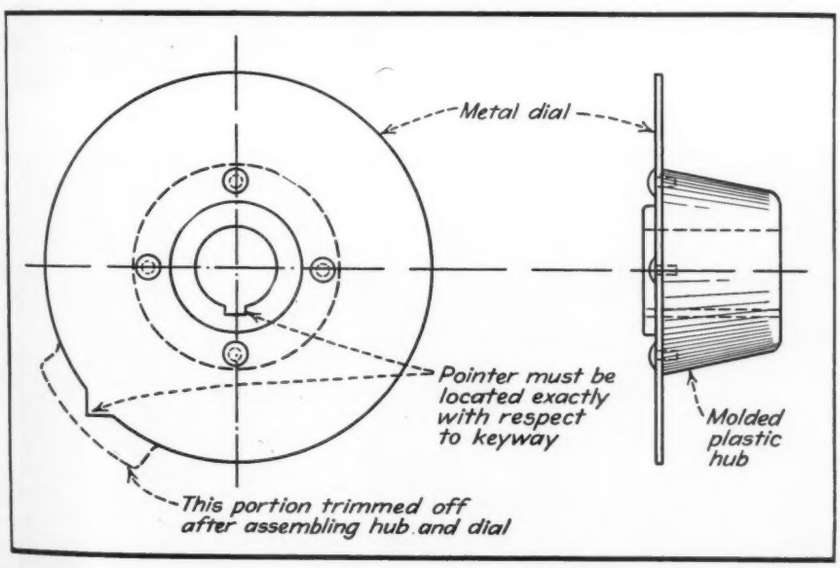


Fig. 5—Actual beam cross-section and assumed cross-section of beam with trapezoidal shape, in which the modulus of elasticity in compression is greater than that in tension. Fig. 6—Actual beam cross-section and assumed cross-section of beam with trapezoidal shape, in which the modulus of elasticity in tension is greater than that in compression. Fig. 7—Actual and assumed effective areas of specimen tested in transverse bending with actual and modified dimensions for calculating stresses

Causes and Cures



Electroplating zinc alloy die castings can cause trouble if not done correctly. The greater the total thickness of the coating, the greater is the protection afforded the base metal and the longer the life of the finish. General Motors Corporation specifies minimum thickness of coating for various grades of plating and different exposure conditions, one of which is illustrated in the photomicrograph above. Chromium coat only 0.000001 in. thick over the nickel is sufficient in this application; thicker coats are often specified.


Metal dial
Pointer must be located exactly with respect to keyway
Molded plastic hub
This portion trimmed off after assembling hub and dial

Extreme accuracy was required in locating the pointer with respect to the keyway in this plastic hub. This called for high accuracy in locating the screw-holes and keyway in the molded piece. Prohibitive variance crept into the molded hubs, caused principally by removal and faulty replacement of cores in the molds. Cure was to blank the dial with surplus stock around the pointer as shown. After assembling, the hub is accurately positioned by its keyway in a trimming die where the pointer is trimmed.

Creep, the effects of which are shown here, always must be taken into consideration when planning lead roofs and linings, especially where slope is steep and service is long. Sheet lead ¼ in. thick and weighing 16 lb. per sq.ft. was placed on the roof of the Cathedral of St. Maurice at Angers, France, completed in 1274. Lead thickness at the ridge is now 3/32 in. while at the eaves the material has gathered into folds 10 in. high. Modern lead alloys have increased tensile strength and creep resistance, and provide sufficient protection in sheets weighing only 2 lb. per sq.ft.

BRISTLES of the first nylon brushes made for portable brush-plating equipment were 0.010 in. dia. They held the electrolyte well but tended to scratch the thin, non-porous coats of such deposits as silver. A softer brush, obtained by reducing the filament to 0.005 in, proved satisfactory.

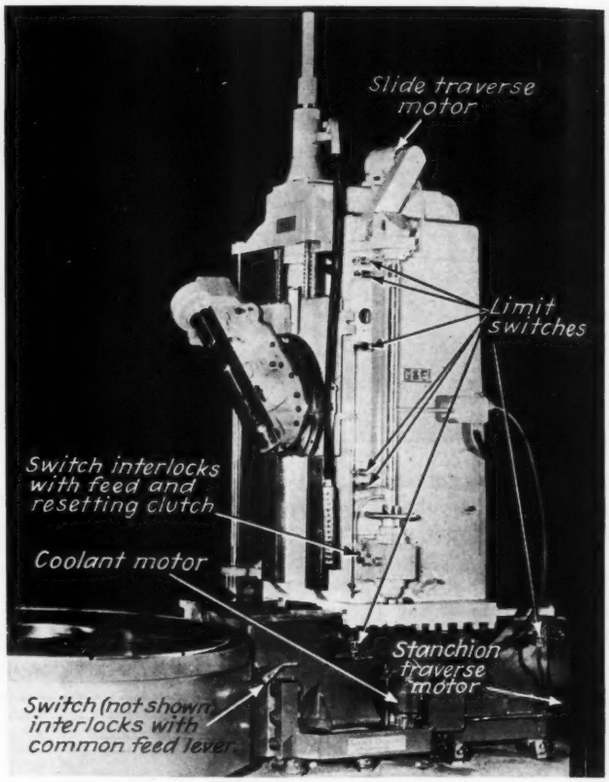
FOR SOLDERING TINPLATE articles to meet government specifications, a commercially pure tin was tried, but cracking was found in the joints after solidification. Trouble was caused by low-melting impurities in the tin, particularly lead, which remained liquid at the grain boundaries after the rest of the metal had solidified. At final stage of solidification, these grain boundaries were pulled apart by shrinkage stresses, forming cracks. Remedy was to substitute a brand of tin with a purity of at least 99.99 percent.

PRODUCT ENGINEERING will pay a minimum of $3 for each example published in Causes and Cures. Where illustrations are necessary, include drawings, rough sketches or photographs.

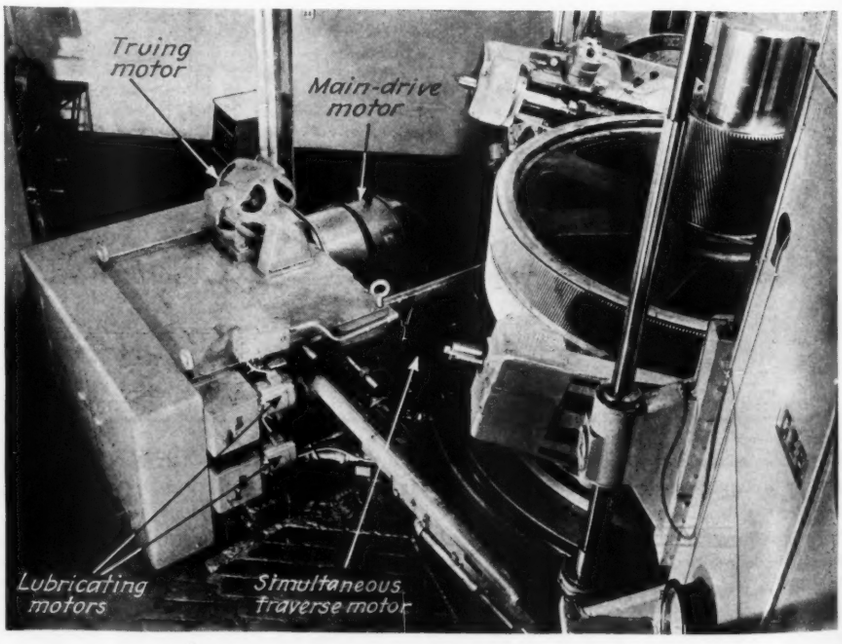
Modern Designs

Electrical Controls for Complex Drives

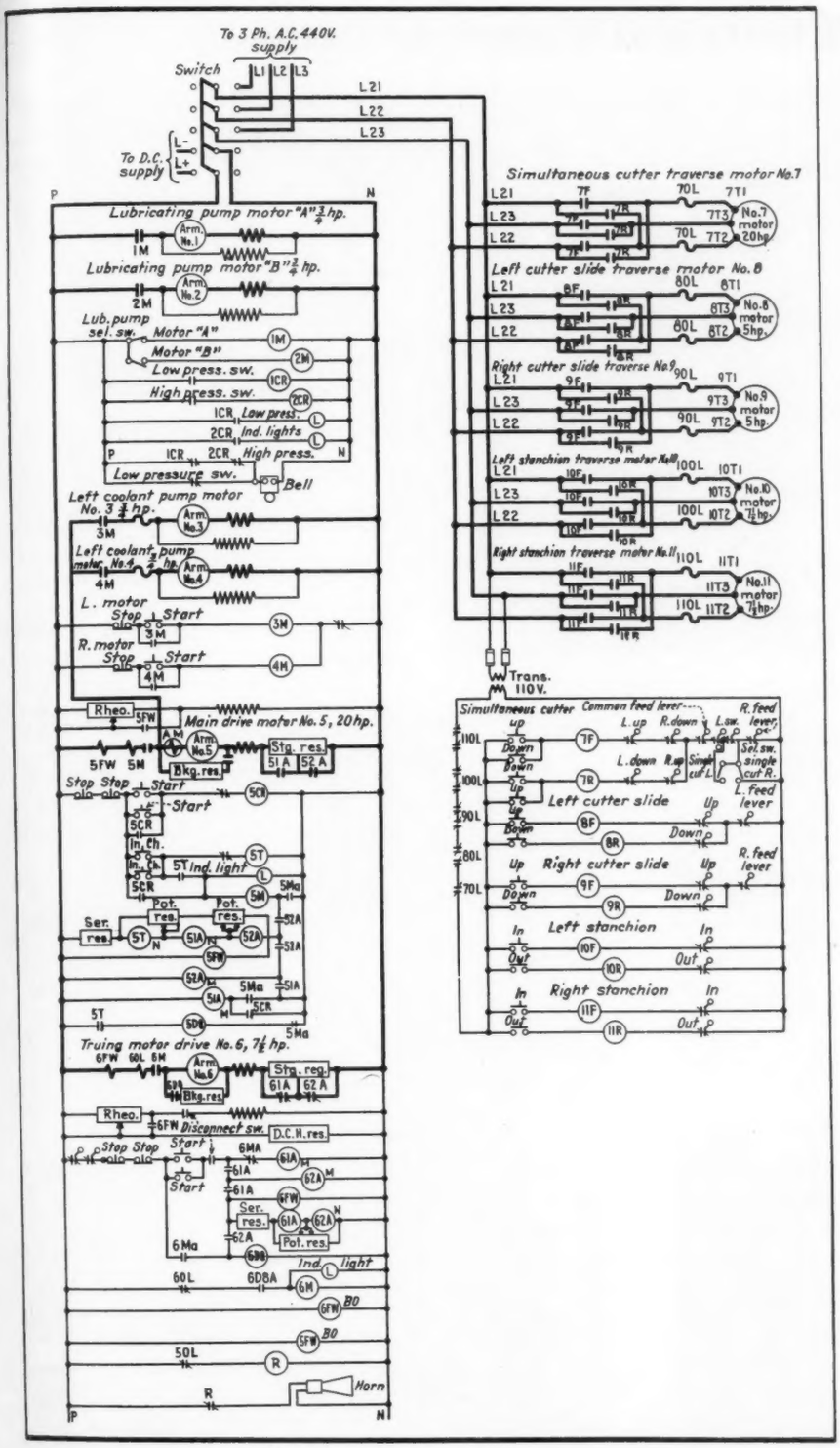
High-speed turbine reduction gear production, potential bottleneck in naval construction for national defense because of the size and accuracy required of the helical gears, has been speeded up by Gould & Eberhardt engineers with a hobber designed to reduce the cutting time almost 50 percent. Two interlocked cutterheads placed opposite each other (one-half of machine shown at right) cuts both right and left-hand portions of double helical gears simultaneously. The necessarily complex operations, the required accuracy and the size of machine involved resulted in the use of certain mechanical features and application of electrical control new to gear cutting machinery.


Slide traverse motor
Limit switches
Switch interlocks with feed and resetting clutch
Coolant motor
Stanchion traverse motor
Switch (not shown) interlocks with common feed lever

Individual motor drives simplify control and transmission problems. Use of a d.c. main drive motor, for example, provides variable hob speeds without gear change boxes. One common differential is used between main drive motor and cutter heads because the heads feed in opposite directions. Stanchion bases, center base, work table, trunnion and the two stanchions are separate Meehanite castings. Having a nominal capacity of 120 in. dia., 48 in. base width at 45 deg. and 2½ diametral pitch and up to 160 in. dia. with gears with low helix angles, the machine weighs 138,000 lb. and requires 20x30 ft. floor space.



Thermal insulation prevents the heat generated by the driving elements and motor from distorting the accurate alignment of the machine. This is particularly essential in a large machine where hobbing large diameter and wide faced gears require several days of continuous cutting to finish one gear. Therefore, the main gear mechanism and driving motors are mounted on a separate gear box connected to the gear cutting machine by splined shafts. All the driving shafts are journaled outside the machine so that the heat will not affect the alignment of the main bases and stanchions. A pump circulates oil continuously through a filter and then to the mechanism in the main gear case, to the work table and to the two index worm-gear sets, and cascades oil over the change gears inclosed in the oil-tight compartment, the lower portions of which form the main reservoir for the lubricating oil.



Eleven motors, six d.c. and five a.c., are used to operate the machine. Flexibility of electrical controls and devices provided a simple solution to the complex problem of coordinating the various drives. D.c. motors for the main drive, lubricant and coolant pump permit the use of storage batteries for stand-by power, thus preventing the machine from accidently stopping during the finish cut, which would leave objectionable marks in the gear teeth and necessitate recutting the gear. Main drive motor is d.c. constant horsepower type, rated at 20 hp. and with four-to-one speed range permits wide selection of hob speeds without change gears. Hob speeds, 18 to 72 r.p.m., are graduated directly on rheostat plates. When motor is used for inching it operates at lowest possible speed regardless of field rheostat setting, being subject to limited dynamic braking. This is not effective when the motor is stopped by the regular stop button because the severe jolt would leave marks in the gears if the hobs were cutting. Thermal-overload protection for the main motor is also admitted for the same reason. The horn signal warns the operator when the motor is overloaded and permits slowing down the motor without bringing it to a stop and without interrupting the cut. Limit switches located on each stanchion serve to stop the main motor when a roughing or finishing cut is completed.

Since the speed of the main motor varies, four separate ¾-hp. motors for lubrication and coolant are used. They are supplied in duplicate with pumps and filters to guard against failure.

The remaining d.c. motor is 7½ hp. variable speed used for rapid rotation of the table when truing the gear blanks. This motor is pushbutton controlled and its field rheostat, mounted on the main panel, is graduated directly in terms of table speed ranging from ½ to 1 r.p.m. Limit switches protect the fine pitch worm from being used in truing.

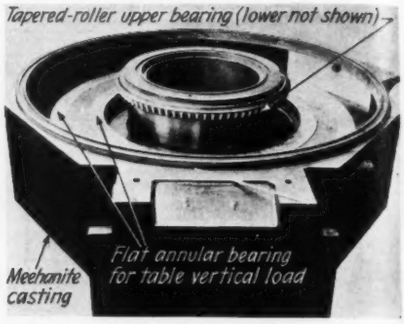
Five a.c. motors are used for traversing: two 7½ hp. for traversing the stanchion; two other 5 hp. motors mounted on top of the stanchion independently position the cutterheads vertically. Before they can be operated, a feed and resetting clutch on each stanchion must be disengaged to disconnect the main speed drive. Interlocks in the control circuit of each motor prevent their operation when the clutch is open. A 20-hp. motor simultaneously traverses both heads to maintain the lead relation to each other and the gear blank. It also rapidly traverses the hobs in opposite direction as in feeding. This motor is interlocked with a common feed lever by a limit switch which cannot be operated unless the feed is disengaged.

A.c. and d.c. circuits are coordinated by the use of a double throw main switch located on the control panel. A neutral position is provided to permit electric inspection or repair. This simple switch eliminates complicated interlock and prevents a.c. motors from running when d.c. motors are in operation and vice versa. Pendant push-button station controls all motors except the lubricating motors. Separate stations arranged with 13 buttons are provided for each cutterhead. Seven buttons on each switch are for dual control of the main, truing, and simultaneous traverse motors. The remaining buttons control the motors pertaining to their respective cutterheads. All controls for the a.c. motors are 110 volts stepped down from the 440 volt power circuit.

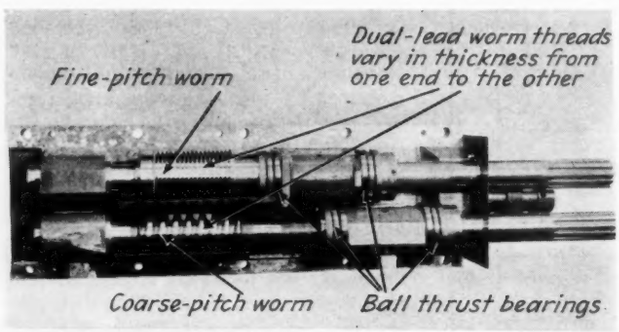
(*Continued on next page*)

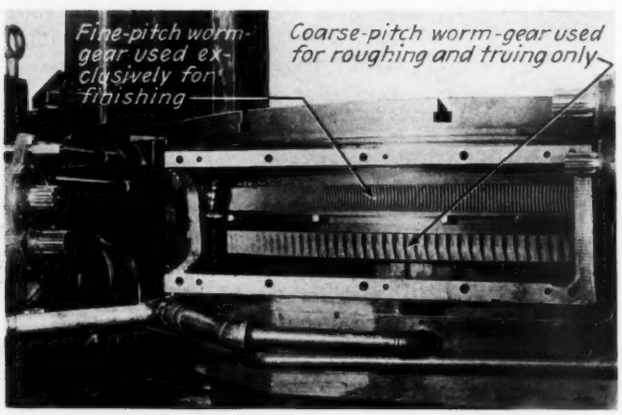
MODERN DESIGNS – Electrical Controls *(continued)*

Heat generated in the electrical controls is kept away from the machine by mounting all motor starters and rheostats on a panel placed in an inclosure in the wall of the air-conditioned room in which the machine is operated. The front of the panel faces inside while the rear is accessible from outside the constant temperature air-conditioned room. All heat generated is dissipated to the outside. Signal lights on the panel board indicate whether the lubricating oil pressure is adequate. In the low-pressure oiling system a bell warns the operators when the pressure reaches a pre-determined minimum. Pressure switches operate both warning light and bell. Hand operated selector switch on the panel brings into operation duplicate lubricating pump because the main motor must not be stopped at any time while machine is engaged in the cutting operation.

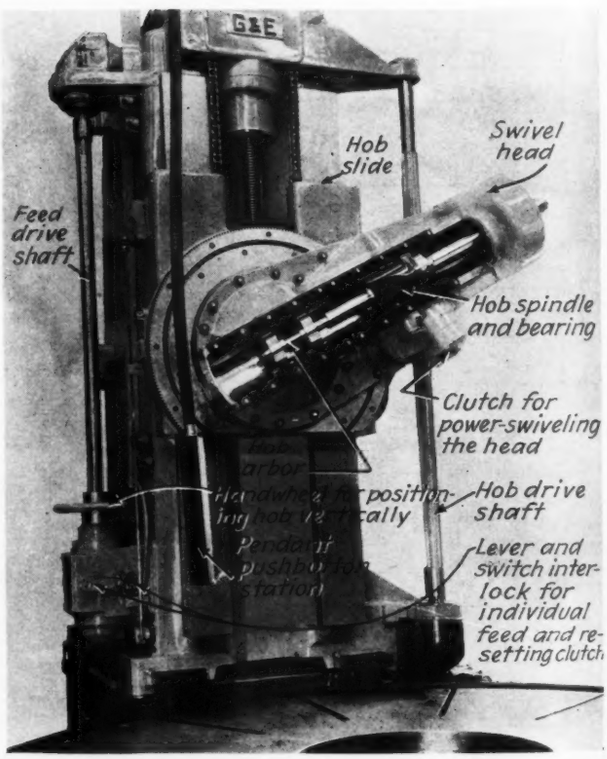


Tapered roller bearings, 54 in. O.D., take the radial thrust and flat annular bearings take the vertical table load. Each bearing race is ground within an accuracy of less than 0.0002 in. run-out between inner and outer surfaces and all rollers made uniform within 0.0001 in. Preliminary tests on the machine showed 0.0003 in. maximum table runout.


Fine-pitch worm
Dual-lead worm threads vary in thickness from one end to the other
Coarse-pitch worm
Ball thrust bearings

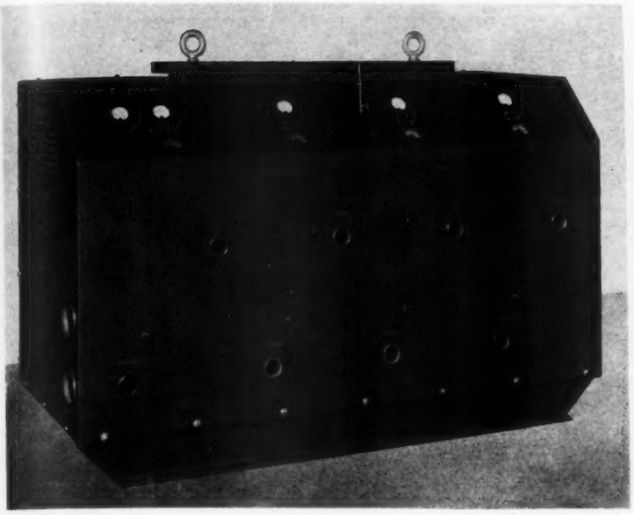


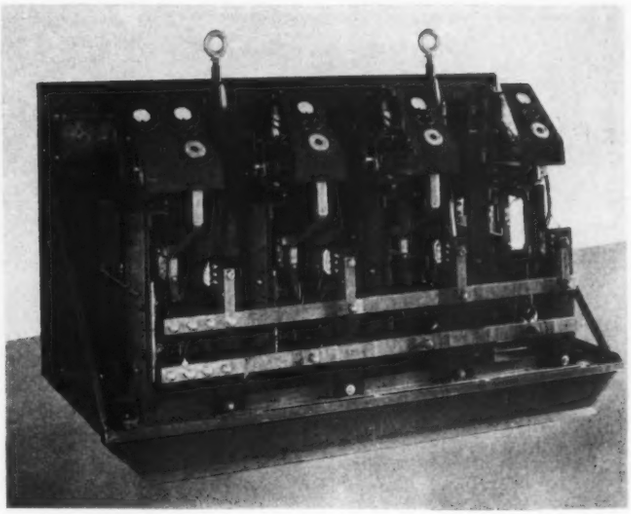
Minimum backlash between the indexing worm and the gear driving the work table and accuracy of the worm gear set are maintained by using a double worm gear, each worm arranged with different lead on each set of profiles, resulting in a thread that varies in thickness from one end to the other. The mating cast Meehanite worm gears are correspondingly modified. Thus the backlash can be controlled by axially adjusting the work without disturbing the center distance and without affecting the transmission of uniform motion. The upper gear set of fine pitch is used exclusively for finish cuts, the coarse pitch for roughing operations.



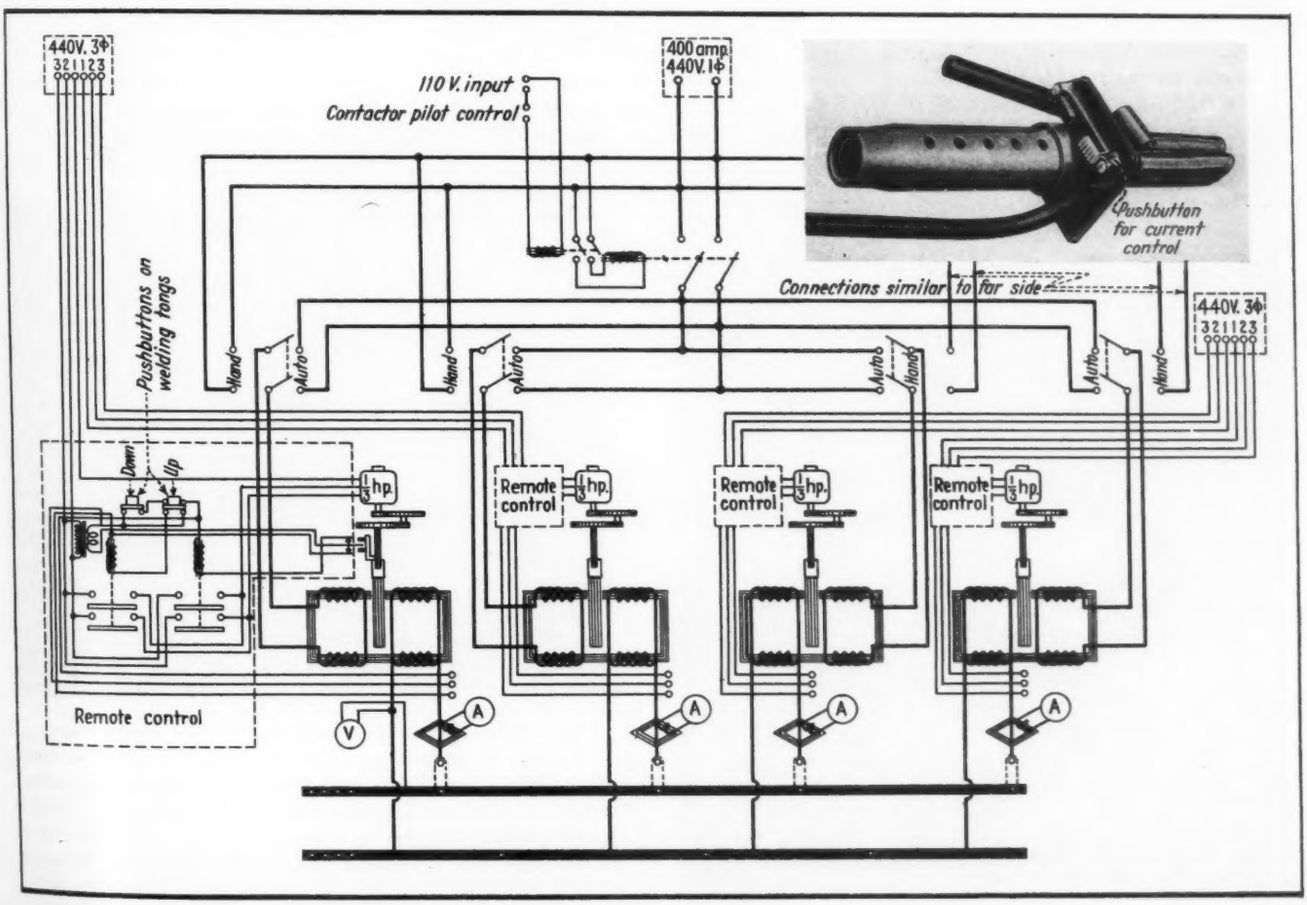
Hob spindles have a reverse taper bearing with the small diameter at the nose end of the spindle to give close support for small diameter hobs. This also permits a larger diameter spindle at the drive end. Spindles are made of high-carbon chrome-manganese-molybdenum steel, heat-treated. Arbors are of similar material, chromium plated to minimize wear and prevent raising of burrs. All drive shafts have multiple splines either with three square keys or ten involute teeth. All main driving shafts and gears are heat-treated, high-carbon chrome-manganese-molybdenum steel. Index worms are case hardened and accurately ground nickel-molybdenum steel. Tapered roller bearings are used for the cutter drives.

Current Control by Flux Diversion

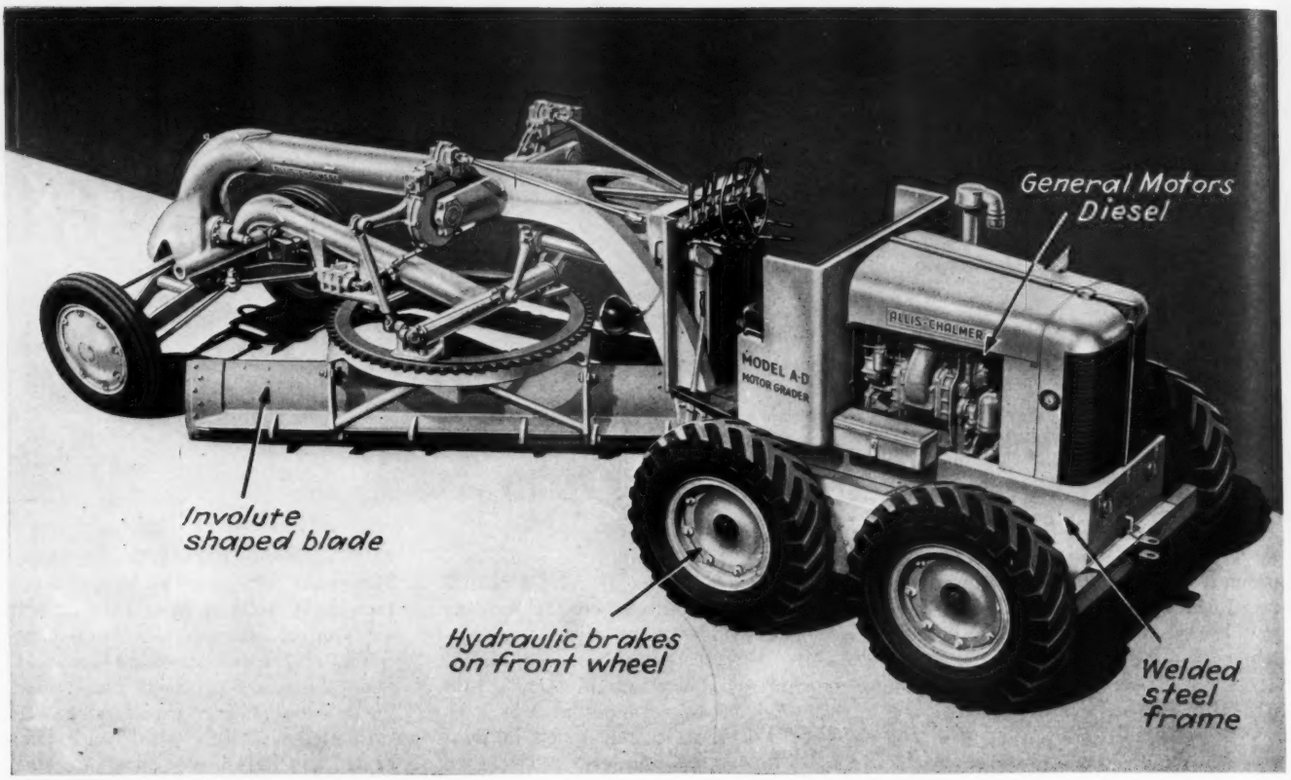




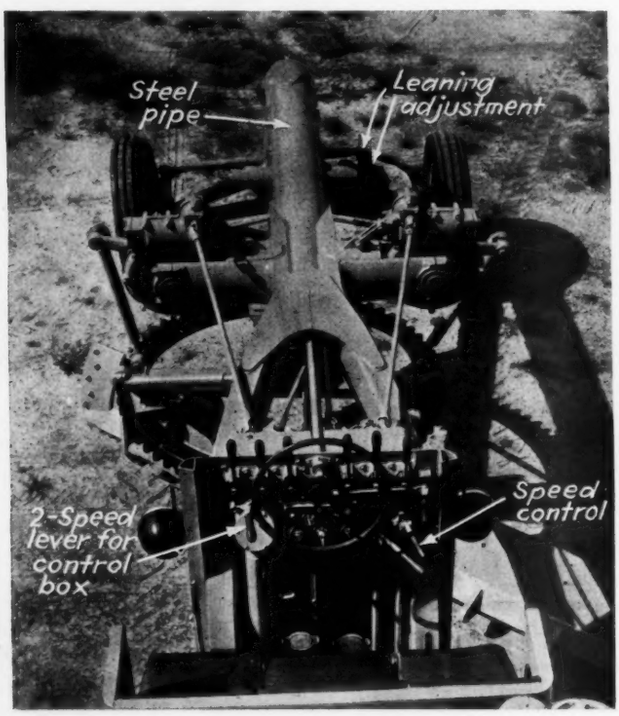
Heavy-duty, multi-purpose Glenn-Roberts welder incorporates a transformer design which controls current by means of flux diversion. An iron core placed between primary and secondary of the transformer is moved vertically by an electric motor and gear train. The position of the core regulates the amount of flux which will be diverted away from the secondary. Accurate positioning by remote control from the pushbuttons on the welding tongs provides "stepless" current control. Circuit permits operation of machine as a single circuit welder with 100 to 2,500 amp. range; as a 2, 3, or 4-operator multi-circuit machine with 2,500 amp. top output; or as four units, each with 100 to 625 amp. output. Maximum voltage which operator contacts is 70 volts or less; only 22 volts on control at tongs. Transformer frame and outer case are grounded. Core plates are precision die stamped for low-loss silicon transformer steel. Insulation is fibre glass and mica. Frame and base are of structural steel.

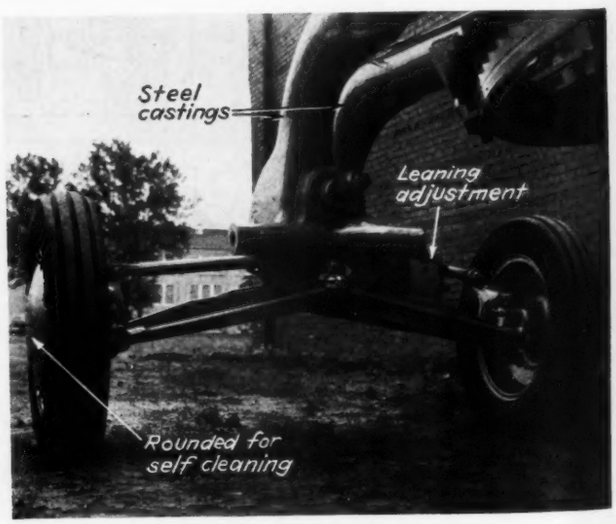


MODERN DESIGNS – Diesel Powered Motor Grader

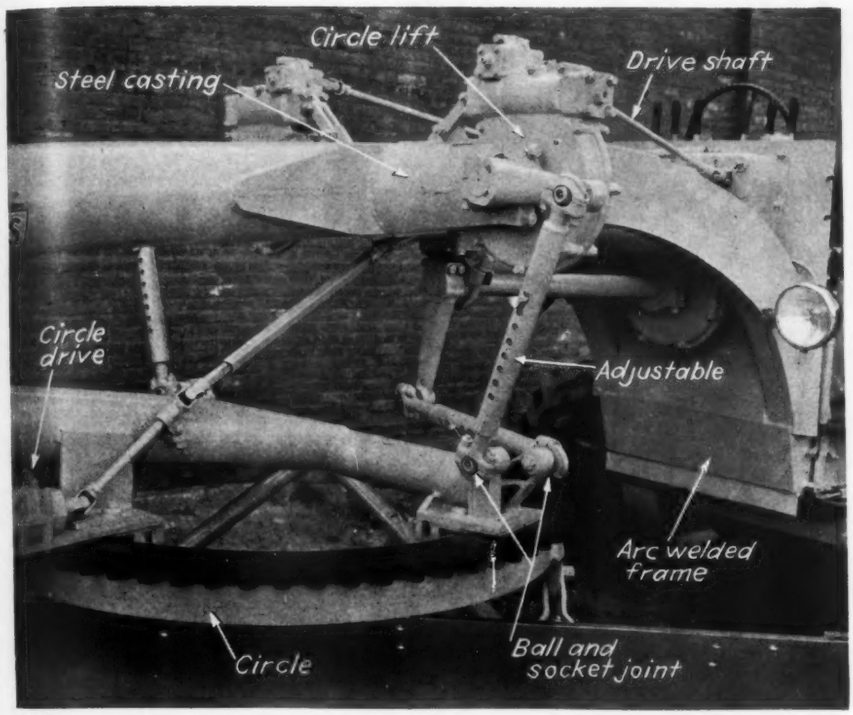


Two-cycle 75 hp. General Motors diesel drives this new Allis-Chalmers grader which features 22 in. clearance at center line of front axle and 6¾ in. between blade and circle, involute curve shaped blade, six forward and three reverse speeds, dual brakes. All-steel welded frame is a tapered beam with maximum section modulus at circle supports where bending moment is greatest. Tubular section of 10 in. extra strong pipe, to resist torsional stresses, is welded to 10 in. structural channels. Channels are also reinforced by tubular members, ends of which are welded to the welds and by steel plate at front of cab. Front post is steel casting as are blade lift arm supports.

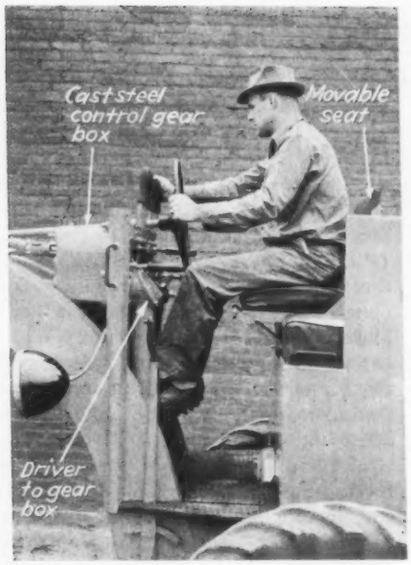




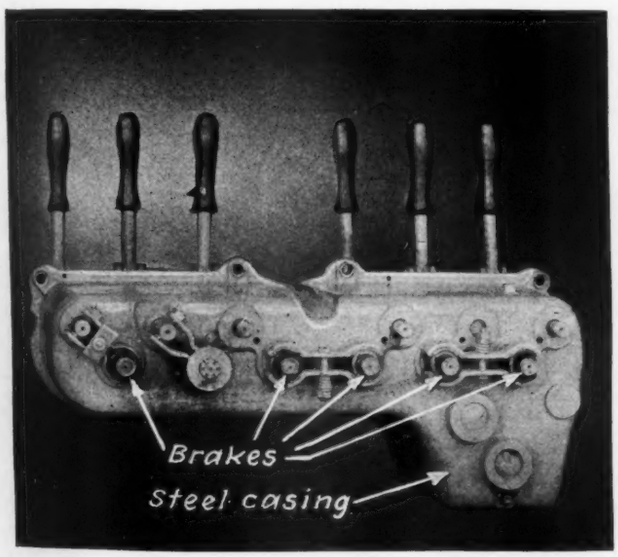
Front axle is welded truss. Center steering control with ball and socket joint permits frame to lean for bank cuts. Worm and worm gear for setting wheels at desired angle are mounted on front axle at right and are driven by power shaft which passes through tubular frame and front post. Draw bar is welded tubular structure with ball and socket end to permit frame to lean.



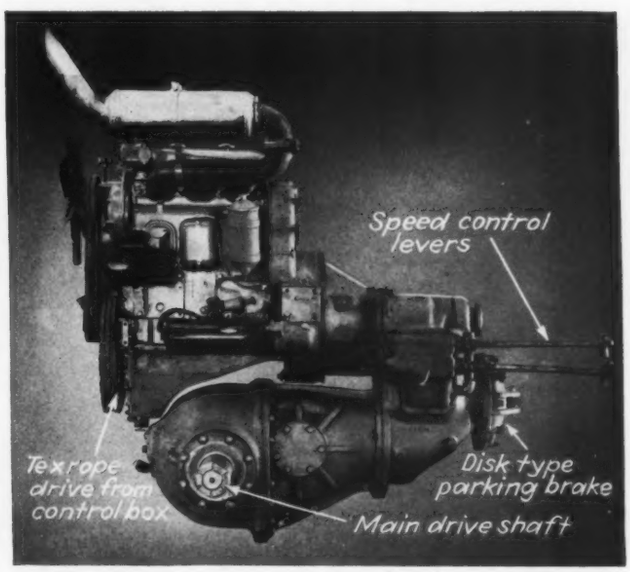
T-shaped welded tubular draw bar forms rigid support for the circle. Circle is made of structural angle, rolled, butt welded and machined on three sides. Gear teeth are hollow steel cast in segments and are welded to the rolled angle. Gear case is mounted out on the frame where it is directly connected to lift arms through short cranks. By using self-locking worm and worm-wheel, the torque holding the blade in operating position is not transmitted to the drive shaft and hence a shaft with high torsional stiffness is not required. Since the shafts only lift the circle they can be of small diameter.



Seat can be moved back and down to level of other cushions when driver stands to operate the machine. Centralized controls include slow or high speed operation of blade and machine position (vertical levers), parking brake, and machine speeds at far right. Electric starter and lights are standard equipment. Transmission controls at driver's right give two ranges of three speeds each (1.48 to 16.6 miles per hour) forward and three speeds in reverse. Lever actuates disk brake on transmission.



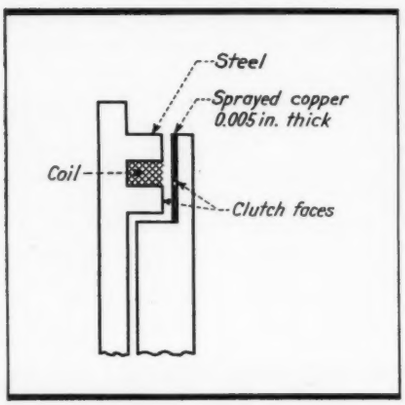
Power from the engine is transmitted to the cast steel control box where it is distributed to spur gears which are connected to the control shafts by bevel faced jaw clutches when levers are operated. Clutches throw out automatically at the limit of motion. Brakes on control shafts are spring loaded. When levers are actuated the bands are forced apart leaving control shafts free to move. When lever is released brakes are automatically applied. Controls can be run at two speeds, slow for close adjustment, rapid for large movements.

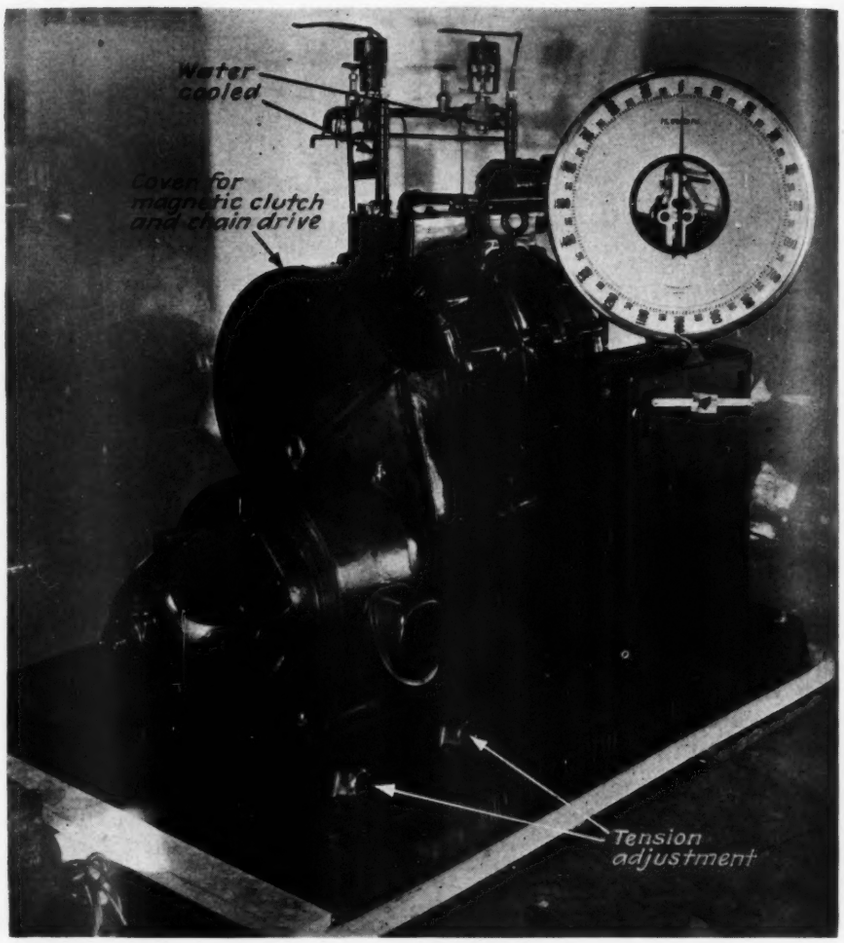


Power plant and transmission unit are designed to have especially short drive shafts to assure rigidity. Disk brake at end of lower gear shaft for parking has Velvatouch brake lining. This brake is also used in combination with the expanding hydraulic brakes on the front tandem wheels in order to prevent entire rear unit from pivoting about main shaft. Motor is mounted on welded channel frame. Drive to four tandem wheels, using welded sprockets and heavy duty chain, is inclosed in a heavy welded case.

MODERN DESIGNS – Aircraft Engine Dynamometer

Designed for running in and testing aircraft engines this Mid-West dynamometer is capable of absorbing 2,000 hp. in a small space by using the Dynamatic eddy current clutch principle described in a previous number (P.E., March 1940). Electric motor, through chain drive and magnetic clutch on dynamometer shaft, runs-in and starts the aircraft engine. When the engine starts, a limit switch cuts the current to the magnetic clutch coil disconnecting the electric motor drive. Since the clutch consists of two alloy steel plates it would be necessary to reverse the current to keep them from sticking together. By spraying one surface with copper the plates separate easily when coil is deenergized. Scale head can be turned in any direction.





Tube type terminal flexible leads used by Westinghouse make possible faster wiring than is possible with the solid tinned copper wire covered with treated cloth tubing. With the solid wire it was necessary to loop the wire for screw connections to major apparatus parts or for coil leads and other small apparatus the connection was soldered. Such screw connections nicked the solid wire, weakening it and often causing breakage. Two or more wires fastened together tend to cut one another in two if tightened too much. With the new type leads, electrical screw connections using No. 4 and No. 10 nickel-plated brass washer head screws are used exclusively, soldering being used only for very small coils. Uniformity of joints is not dependent on the human element. And the cable itself, not the die-formed and stamped terminal connection, is the weakest link in this chain.

Vanco welding process joins S.A.E. 1010 tubing to a 2½-in. dia. steel washer for part of a motor support. A $\frac{3}{16}$-in. rod slides through hole in 0.014 in. thick washer and 0.084 in. tube. Weld is made with little flash at an average rate of 1,200 pieces per hour.

CRANK MECHANISM MOTIONS

New Methods for Their Exact Determination—I

Dr. WERNER F. VOGEL
Wayne University, Engineering College

THE BASIC MECHANISM of all engines, pumps and compressors is the crank-connecting rod linkage. Machine tools, packaging machinery and many other machines are further examples of the widespread application of this important mechanism. The engineering problems in its design deal fundamentally with the masses and accelerations and the resulting inertia forces created. Factors that determine the magnitude of the forces or stresses developed include the mass of the connecting rod, the ratio of crank radius r to rod length l, expressed as $\lambda = r/l$, and the magnitude of the accelerations, which for a given mechanism increase with the square of the speed of rotation. Obviously, the higher speeds for which modern machinery must be designed require the engineer to give more thorough consideration and study to the calculation of the forces created in the crank-connecting rod mechanism. Indeed, the approximate methods generally used can be easily replaced by more accurate procedures for analyzing and calculating the acceleration forces.

A characteristic feature in the development of the modern crank mechanism has been the steadily increasing value of $\lambda = r/l$. One reason for this is that thereby the value of l is reduced and, therefore, the mass of the connecting rod is likewise reduced. A second reason for selecting the value of λ as great as possible is that it results in a more compact mechanism. But mass is only one of the two multiplying factors which determine the inertia forces. The other factor, the acceleration, is also influenced by an increasing value of λ and at extremely great values of λ, the maximum possible value being 1, (Ref. 1), the accelerations will be appreciably greater than the maximum values determined by the usual approximate methods, as will be demonstrated. In *Table 1* are given r/l values commonly used.

Table 1—Usual Ratios of Connecting Rods

Engine	$\lambda = r/l$
Steam Engines	
Locomotives	0.143–0.167
Stationary Engines	0.2
Steamships	0.222–0.25
Gas and Diesel Engines	
Trucks	0.222–0.232
Passenger Cars	0.238–0.278
Airplanes	0.278–0.313

This series of articles will present for the first time new exact equations and graphical methods for the motion of the crank mechanism, equations, which for certain problems are even simpler to compute or construct than the conventional approximate formulas. Moreover, the equations permit the use of new graphical methods on an exact basis. And for the first time there is presented such a definite general algebraic solution for the maximum velocity

Table II—Principal Equations of Motion of Crank Mechanism

Value for Piston	Symbol	Exact Equation	No.	Approximate Equation	No.
Displacement	x	$x = r\,(1-\cos\alpha) + l\,(1-\cos\beta)$ where in: $\sin\beta = \lambda \sin\alpha$	(1)	$x = r\left[(1-\cos\alpha) + \frac{\lambda}{2}\sin^2\alpha\right]$	(A1) and (A2)
Velocity	$c = \frac{dx}{dt} = \frac{dx}{d\alpha}\frac{d\alpha}{dt}$	$c = r\,\omega\,\frac{\sin(\alpha+\beta)}{\cos\beta}$	(3)	$c = r\,\omega\,(\sin\alpha + \frac{\lambda}{2}\sin 2\alpha)$	(A3)
Acceleration	$a = \frac{dc}{dt} = \frac{dc}{d\alpha}\frac{d\alpha}{dt}$	$a = r\,\omega^2\left(\frac{\cos(\alpha+\beta)}{\cos\beta} + \lambda\frac{\cos^2\alpha}{\cos^3\beta}\right)$	(4)	$a = r\omega^2\,(\cos\alpha + \lambda\cos 2\alpha)$	(A4)
Increase of Acceleration	$j = \frac{da}{dt} = \frac{da}{d\alpha}\frac{d\alpha}{dt}$	$j = -c\,\omega^2\left[1 + \frac{3\,(1-\lambda^2)\,\lambda\cos\alpha}{\cos^4\beta\,(\cos\beta + \lambda\cos\alpha)}\right]$	(5)	$j = -r\,\omega^2\,(\sin\alpha + 2\,\lambda\sin 2\alpha)$	(A5)
Slope of $a-x$ Curve	$\tan\vartheta = \frac{da}{dx} = \frac{j}{c}$	$\tan\vartheta = -\omega^2\left[1+\frac{3\,(1-\lambda^2)\,\lambda\cos\alpha}{\cos^4\beta\,(\cos\beta+\lambda\cos\alpha)}\right]$	(6)	$\tan\vartheta$ parabola $= -\omega^2\frac{1+4\lambda\cos\alpha}{1+\cos\alpha}$	(A6)

The equations for x, a and c are called Principal Equations of the Crank Mechanism.
For significance of r, l, λ, α, and β, see Fig. 1 and tabulation above.

Approximate Equations are designated by an A preceding the equation number.
For error of the approximations, see Figs. 5 to 9.

$\omega = \frac{\pi\,n}{30}$ is the constant angular velocity of the crank.

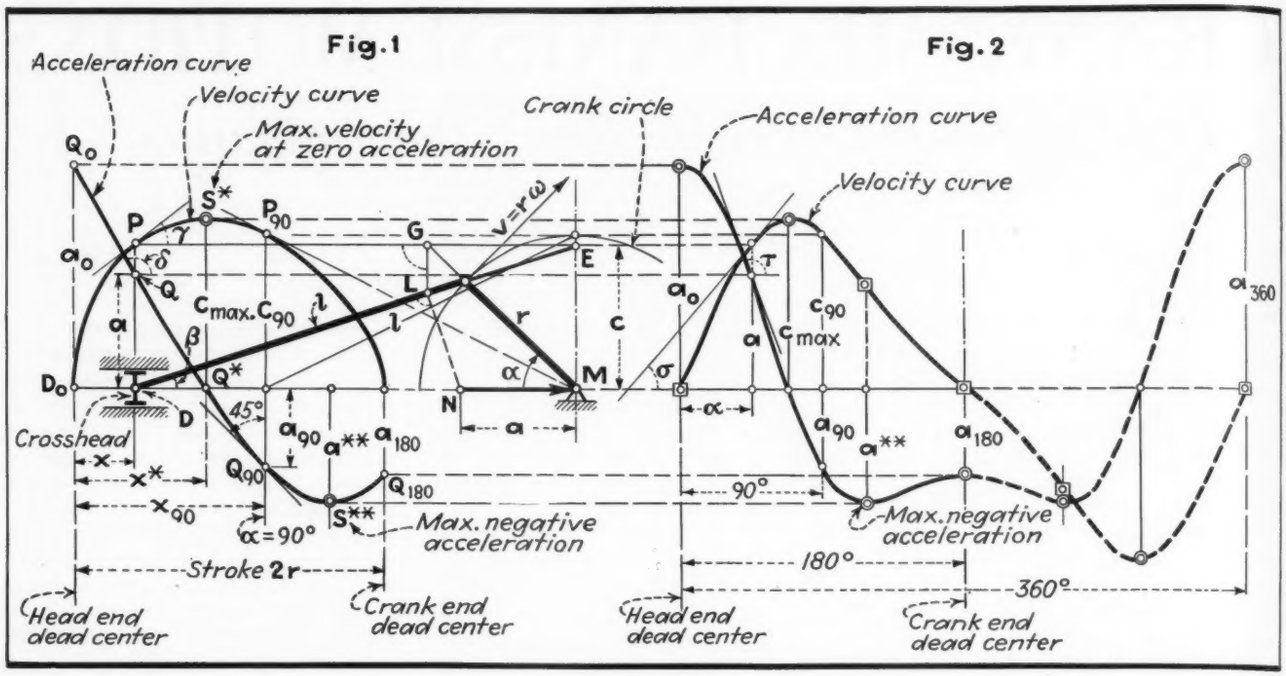


Fig. 1—Piston accelerations and velocities plotted against crosshead displacements. Numerical subscripts refer to angular position of the crank from head-end dead center. To construct the velocity curve, for any position of the crank extend the line of the rod to intersect the vertical ME. Project E horizontally to the left till it intersects the vertical DP passing through the crosshead pin. P will be a point on the velocity curve. To construct the acceleration curve (Ref. 4) the crank is extended to intersect the line PE at G; GL is perpendicular to GE and LN is perpendicular to the line of the rod. NM is the acceleration a of crosshead at position x distance from head-end dead center. The two shaded areas are equal.

*Fig. 2—Piston velocities and accelerations plotted against angular displacement of the crank. Inflection points in the solid portion of the velocity curve occur only at the three vertices in the acceleration curve. Maximum negative acceleration is the only one of these three vertices, which appears also as a vertex with horizontal tangent, namely S**, in the acceleration curve of Fig. 1. The crank's angular displacement α represents also the time t. The acceleration curve is symmetrical about the vertical axis representing the crank end dead center. The velocity curve from 180 to 360 deg. is the reflection of the curve from zero to 180 deg. The scale for a is selected arbitrarily and points are projected over from the corresponding points in Fig. 1 as indicated by dash lines*

of the piston, that it is simple enough for practical numerical evaluation instead of commonly used approximations.

The following symbols, as shown in Fig. 1, will be used throughout these articles:

x = piston or crosshead displacement from head end dead center
c = piston or crosshead velocity
a = acceleration of piston or crosshead
ω = angular velocity of crank
r = crank radius
l = connecting rod length
$\lambda = r/l$
α = arbitrary angular displacement of crank, equals zero when piston is at head end dead center
β = connecting rod angle
$\tan \gamma$ = slope of velocity-displacement curve of piston
$\tan \delta$ = slope of acceleration-displacement curve of piston

Equations of motion. The motion of the crosshead and piston is expressed by the principal equations of motion of the crank mechanism; they are for the purpose of computing the displacement x, the velocity c, and the acceleration a of the crosshead. Since crosshead and piston have the same motion, henceforth we shall refer to it only as that of the piston. According to symmetrical relations of the lines of motion, as shown in Figs. 1 and 2, it is not necessary to investigate the return stroke of the piston. Therefore, all equations presented refer only to the motion within the range from 0 to 180 deg. inclusive.

Exact principal equations have been published for some time, but, unfortunately, they have not become as well known in engineering circles as it would seem their significance demanded.

Exact principal equations. These are shown in the accompanying *Table II*. Equations (1) to (4) of the principal equations can be found in a publication by Pinzger (Ref. 2). They allow a direct computation of x, c and a if the position of the mechanism is given by the angular displacement of the crank, but can be applied to the range from 180 to 360 deg.

Equations (1) and (2) can be found by geometric conclusion from Fig. 1. Equations (3) and (4) are obtained by simple or double differentiation of Equation (2) or by other procedures of dynamics, for instance, by means of the instantaneous center and by methods of relative motion. They are valid under the assumption, in most cases admissible, that the number of revolutions per minute, n, and therefore the angular velocity of the crank, $\omega = \pi n/30$, is constant.

Equations (5) and (6) of Table II are formulas for the increase of the piston acceleration and for the slope of the true curve of acceleration, shown in Fig. 1. They were derived by the author for purposes of more thorough research (Ref. 3).

From Equation (2) the formulas of *Table III* were obtained by differentiation. Their application will be shown in the discussion of the motion of the connecting rod.

By computing x, c, and a for different crank angles α the acceleration and velocity curves can be plotted against x or α, as shown in Figs. 1 and 2. However, because of its involved character, few engineers have felt Equation (4) adaptable for practical computation; rather they prefer simpler approximate formulas.

In these articles, approximate equations will be indicated by an *A* preceding the equation number.

Approximate forms of the principal equations. The approximate principal equations cited in engineering manuals are tabulated as Equations (A1) to (A4) at the right side of Table II. Equation (A5) is valid for the increase of the approximate acceleration defined by Equation (A4).

The computation of x, c and a by aid of the approximate Equations (A1) to (A4) is especially simple when the angular displacement of the crank is given or selected. If, for different crank positions, we trace the so computed different values of the acceleration a over the displacement x of the piston, we get a parabola as the approximate curve of acceleration, as shown in Figs. 3 and 5. This parabola has entered into engineering literature as Radinger's Parabola of Acceleration or simply as Parabola of Acceleration. Fig. 3 shows a convenient graphical construction of this parabola, the slope of which, see Fig. 5, can be expressed by Equation (A6) as tabulated.

In many practical cases Radinger's parabola may be adopted as a sufficiently exact substitute of the true curve of acceleration. It is regrettable that many engineering handbooks do not even mention that Radinger's parabola is only an approximation and none of them discuss the errors involved in applying it.

In order to enable everyone to decide for himself, whether in his case this approximation is permissible, the author has made a thorough investigation of the maximum errors of the approximate forms of the principal equations, and has given exact equations for computing and analyzing these errors (Ref. 3). The results, as shown by the curves of the greatest errors in Figs. 7 to 9, can be summarized as follows:

1. Displacement of Piston Approximate Equations (A1) and (A2): The error at the dead centers equals zero. The greatest error always appears in the perpendicular position of the crank, $\alpha = 90$ deg. It grows, as shown in Fig. 7, only slowly with increasing λ and is negligible for values of λ between 0 and 0.3, which is the most frequently used range.

2. Piston Velocity, Approximate Equation (A3). The error is zero at the dead centers and in the perpendicular position of the crank, $\alpha = 90$ deg. The error has two vertices; they appear in those crank positions which correspond to the two intersecting points A and A' in Fig. 5. The vertex error corresponding to A' is always the greater one; for this, the α-curve and the greatest error-curve in Fig. 8 are shown.

The greatest error for c is negligible in the usual range of $\lambda = 0$ to $\lambda = 0.3$, but as shown in Fig. 8, it then increases greatly with increasing λ values.

(Continued on next page)

Table III—Equations for the Angular Motion of Connecting Rod

	Symbol	Equation	No.
Connecting Rod Angular Displacement.....	β	$\sin \beta = \lambda \sin \alpha$	(2)
Connecting Rod Angular Velocity.....	$\omega_\beta = \frac{-d\beta}{dt}$	$\omega_\beta = -\omega \lambda \frac{\cos \alpha}{\cos \beta}$	(7)
		$= -\omega \operatorname{cotan} \alpha \tan \beta$	(8)
Connecting Rod Angular Acceleration.....	$\epsilon_\beta = \frac{d\omega_\beta}{dt}$	$\epsilon_\beta = (\omega^2 - \omega_\beta^2) \tan \beta$	(9)

The positive sign signifies clockwise rotation like that of the crank.

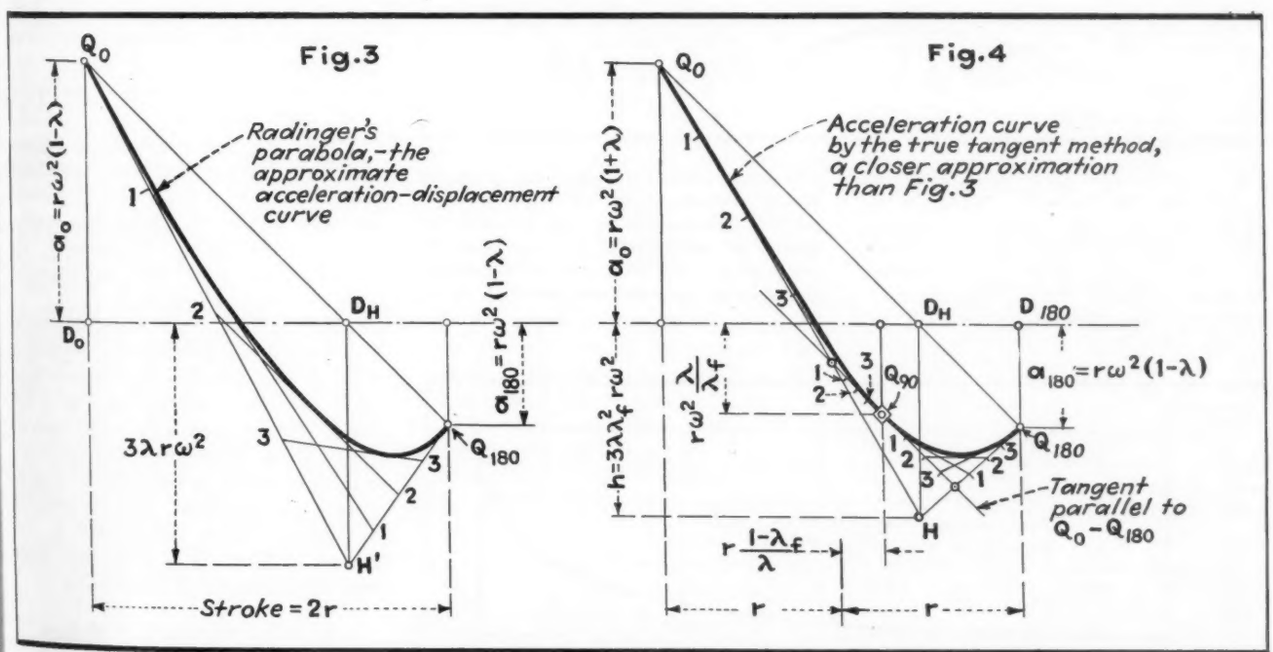


Fig.3
Radinger's parabola,-the approximate acceleration-displacement curve
Stroke = 2r
Fig.4
Acceleration curve by the true tangent method, a closer approximation than Fig.3
Tangent parallel to Q0-Q180

Fig. 3—To construct Radinger's parabola. Using arbitrary scales, locate points Q_0 and Q_{180}, which determines point D_H. This with the calculated value of $3\lambda r\omega^2$ locates the point H'. The rays $H'Q_0$ and $H'Q_{180}$ are each divided into equal segments, the corresponding points connected and the Radinger Parabola is drawn tangent to the connecting lines. Fig. 4—True tangent method for constructing the acceleration curve, suitable for all values of λ less than 0.4 Points Q_0, Q_{90}, Q_{180} and H are located, calculating λ_f from the relationship $\lambda_f = \sqrt{1-\lambda^2}$. The two sets of points 1-1, 2-2, and 3-3 are connected and the two curves Q_0Q_{90} and $Q_{90}Q_{180}$ are drawn tangent to these connecting lines

3. Acceleration of Piston, Approximate Equation (A4). Fig. 5 shows impressively how much Radinger's parabola can deviate from the true line of acceleration. Error of acceleration is zero at all points where the curve of acceleration intersects the parabola; the error is zero not only at the dead centers Q_0 and Q_{180} ($\alpha = 0$ deg. and $\alpha = 180$ deg.), but also at the points A and A', which positions, from the previous discussion of approximate Equation (A3), are already known.

In Fig. 6 we find three vertices of errors S_1, S_2, and S_3, of which S_2 is the most important. It is independent of λ and occurs always at $\alpha = 90$ deg., the perpendicular position of the crank.

Of the other vertices S_1 and S_3 the latter is the more important since here the error is always greater. The crank position corresponding to S_3 may be read from the dotted curve in Fig. 9, as indicated thereon. The two error-curves of the vertices S_3 and S_2 intersect each other in Fig. 9. From $\lambda = 0$ to $\lambda = 0.75$, S_2 gives the greatest error of acceleration; from $\lambda = 0.75$ to $\lambda = 1$, the S_3 curve, however, indicates the greatest errors. Thus the greatest value of the acceleration's error always follows the composed curve which is determined by the greater ordinates of the intersecting curves.

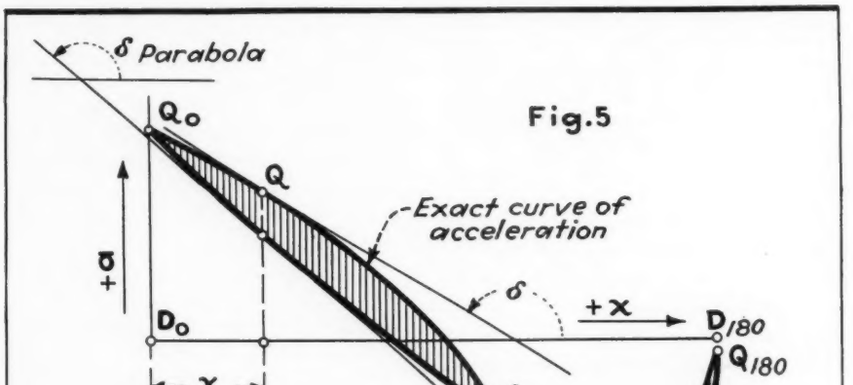


Fig. 5—The exact curve of acceleration and Radinger's Parabola. Location of points of zero error, A and A' varying with λ, also determines the crank position for maximum error of the velocity as given in Fig. 8. Fig. 6—The curve of errors is obtained directly from Fig. 5. The error vertex S_2 occurs always at the 90-deg. position of the crank ($\alpha_2 = 90°$) and is independent of the location of maximum acceleration $S_{parabola}$ and S. See Figs. 7 and 9 for curves of maximum errors in the approximate principal equations

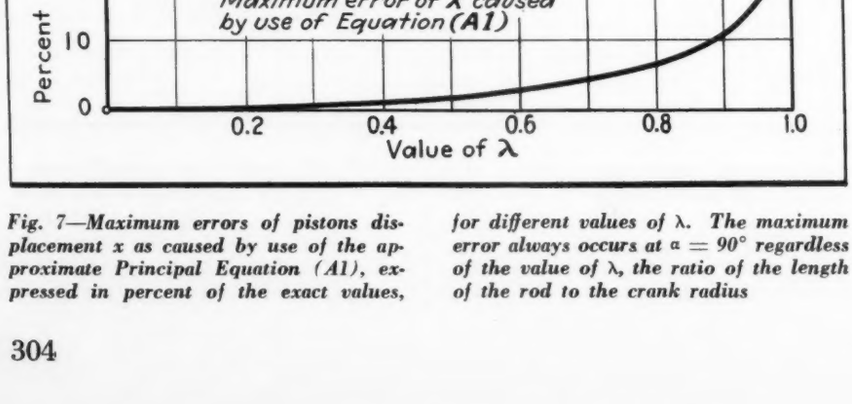


Fig. 7—Maximum errors of pistons displacement x as caused by use of the approximate Principal Equation (A1), expressed in percent of the exact values, for different values of λ. The maximum error always occurs at $\alpha = 90°$ regardless of the value of λ, the ratio of the length of the rod to the crank radius

4. Errors of Radinger's Parabola. The errors of the abscissas and the ordinates of the Radinger's parabola have been discussed. Except for the rarely used range λ greater than 0.75, where the application of Radinger's parabola ought to be excluded anyway, for both abscissa and ordinate the greatest errors occur at the 90 deg. crank position. The outer tangents of Radinger's parabola at Q_0 and Q_{180}—often utilized for a graphical construction of this curve, as in Fig. 3—deviate from those of the true curve of acceleration.

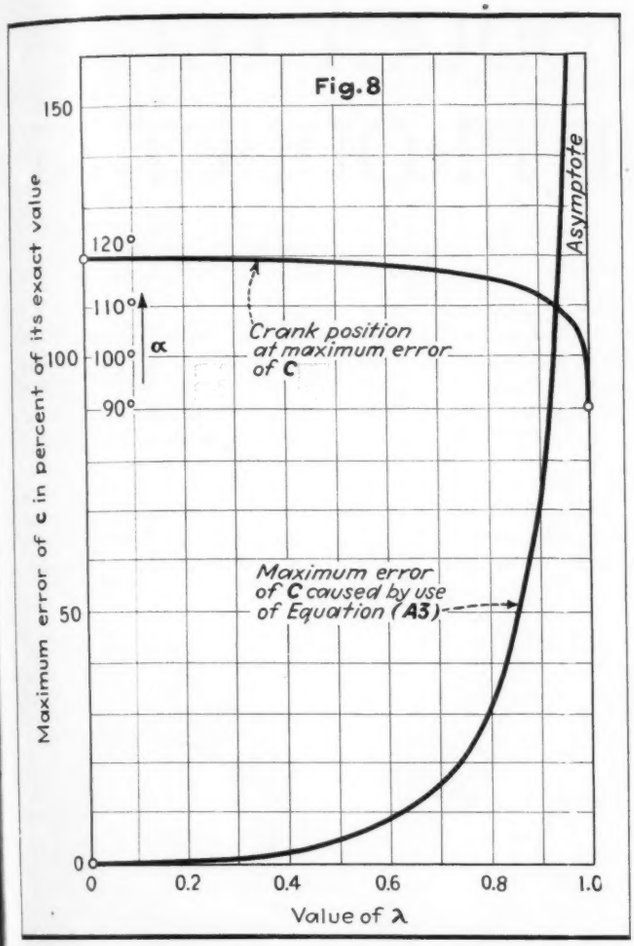
All this is eliminated in the new approximate acceleration-displacement curve, shown for the first time in Fig. 4. This curve is exact at Q_{90}, just where the Radinger's parabola has its greatest deviations. The three tangents at Q_0, Q_{90} and Q_{180} are also exact. Therefore, the author expects his curve to be much closer to the exact one than is the Radinger's parabola.

Of course, this expectation is justified only for a reasonable range, such as between $\lambda = 0$ and $\lambda = 0.39$, wherein the true curve of acceleration has no point of inflection. Fortunately, the limiting value $\lambda = 0.39$ is much greater than the usual λ values.

For λ greater than 0.4 both of the approximate constructions shown in Figs. 3 and 4 should be avoided. However, the true tangents of Fig. 4 will prove helpful even within this extreme range when the construction of the acceleration curve is aided by locating its points of inflection and its negative maximum. These points will be discussed in Part III of this series which will appear in the August issue.

Use of Numerical Tables

If, in the principal equations, r and ω are each put equal to unity, one gets the equations for dimensionless unit crank gear. These signify that the motion diagrams of all crank gearings with the same ratio λ are geometrically



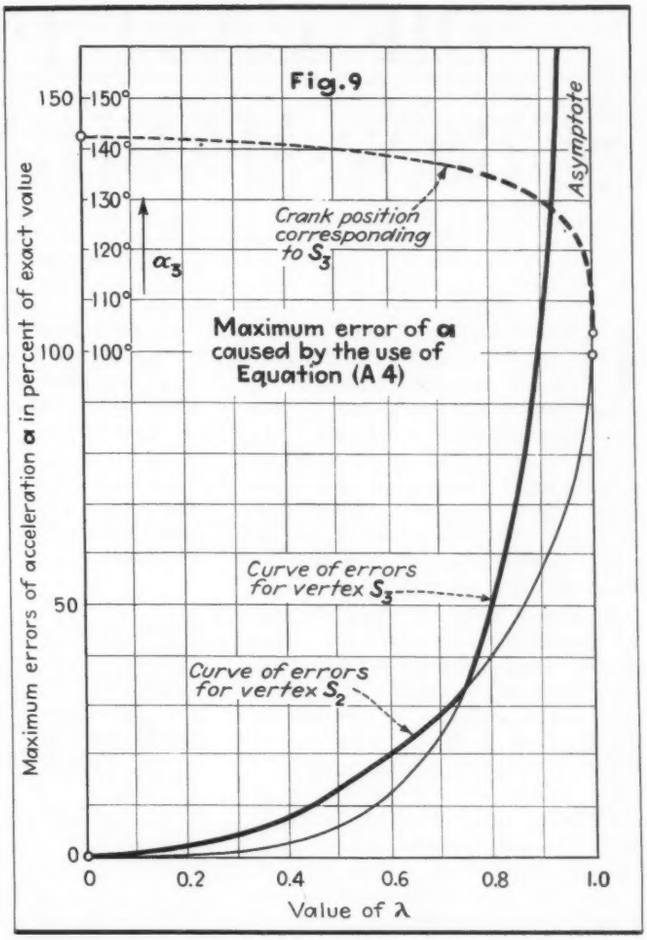


Fig. 8—Maximum error of piston velocity c caused by application of approximate principal equation (A3) corresponds always to the crank position α of point A' in Fig. 5, which varies with the value of λ. Fig. 9—Maximum error of the acceleration a when a is computed by use of the approximate principal equation (A4). The error is plotted in this chart as the difference between the exact value and the approximate value as calculated by Equation (A4), divided by the exact value and expressed as a percentage. The maximum error follows the heavy curve corresponding to the error vertices S_1 and S_3 of Fig. 6. It should be noted that vertex S_1 leads to an error curve, not shown here, with smaller ordinates than the heavy maximum error curve shown here. Error vertex S_2 always occurs at the 90 deg. position of the crank, that is, $\alpha_2 = 90$ deg., while the crank position α_3 for the vertex S_3 can be read from the dotted curve

similar. One needs, therefore, only to tabulate the values for the unit crank gearing of a certain λ value in a numerical table in order to obtain at once the proportional values for every other crank gearing of the same ratio of rod length to crank radius. For a crank mechanism of a given crank radius r and a given crank angular velocity ω every wanted value is then k times as great as the corresponding tabulated value of the unit mechanism. Herein k is a proportionality factor, having the following fixed values:

For all angular terms (angles, angular velocities and angular accelerations) ..... $k = 1$
For displacement of piston ..... $k = r$
For velocity of piston ..... $k = r\omega$
For acceleration of piston ..... $k = r\omega^2$

Obviously, for a fixed λ value, we need only one set of curves of motion, namely, that of the unit crank mechanism, for all possible applications. The variability of r and ω only influences the scales which can be easily determined by the above given k values.

Because of the extraordinary simplifications which can be obtained in this way, most of the engineering manuals do contain such numerical tables for α, x, c, and a at different λ values. But, unfortunately, almost all of these numerical tables are computed on the basis of the approximate main equations.

Editor's Note—In the second installment of this series of four articles, the author will present new simplified exact methods for the arithmetical computation and graphical construction of the crank-connecting rod motions. Two new graphical constructions of the acceleration curve which enable locating dead center acceleration will be shown along with suggestions to simplify plotting the curve by means of tangents.

REFERENCES

(Ref. 1) For practical applications of crank mechanism with λ-values much greater than usual, even with $r = l$, see Rittershaus, Zur Konstruktion der Beschleunigunskurve eines Kurbelmechanismus. Zeitschr. des Vereins Deutscher Ingenieure 1883, pg. 283.

(Ref. 2) Pinzger, Zeitschrift des Vereins Deutscher Ingenieure, Berlin 1883, pg. 137.

(Ref. 3) W. Vogel, Der Einfluss des Schubstangenverhältnisses auf die Bewegungsvorgänge beim Kurbeltrieb. Automobiltechnische Zeitschrift, Berlin, 1937, pg. 336. Several of the figures in the P. E. article are taken from this publication.

(Ref. 4) Mohr, Die geometrische Konstruktion der Beschleunigungen der ebenen Bewegung. "Der Zivilingenieur" 1879, pg. 613.

DIE-CAST PRODUCT OR SCREW MACHINE PRODUCT—I

HERBERT CHASE

DESIGNERS of metal products are aware of the facility with which parts can be made by the screw machine and by die-casting. It is less generally realized, however, that certain types of parts can be manufactured alternatively by either type of machine. Both types of product have advantages and limitations which deserve consideration when making a choice between the two.

As distinct from the "hand" or non-automatic type, the automatic screw machine is an automatic hollow-spindle turret lathe and, within size limitations, can produce rapidly and economically almost any shape of part which can be turned from bar or tubular stock by a lathe. By the use of tools, fixtures or attachments, some of which are small machines, many supplementary operations such as slotting, milling of flats, burring and cross drilling can be performed. These and other operations are sometimes done more advantageously on separate secondary machines. Operations which the screw machine proper can do on the cut-off end of the piece are somewhat limited, however, some machines are provided with attachments for removing the burr and for doing other light operations after the piece has been cut off from the work stock and transferred to the attachment.

It is also possible, when equipped with a magazine feed, to use a screw machine as a chucking machine. With a magazine feed pieces already partly machined in prior operations, on the same or a duplicate machine, are loaded into the spindle collet and then machined, generally on the cut-off end, much as if the machine were using bar or tubular stock, except for the method of feeding.

Parts produced by die-casting follow an entirely different cycle. Molten metal is fed under high pressure into a steel die and solidifies quickly; the casting is ejected almost immediately. The die may have a single or several cavities, all the same or all different as required; or several separate dies can be installed in the machine. After the casting operation the parts must be cut or broken from the gate, and the flash be sheared or otherwise machined off.

In Table I are listed the factors that are usually given consideration when determining whether a product should be made by die-casting or on a screw machine.

MATERIALS. The alloys used to the greatest extent for die-castings are those based on zinc. Next in importance are those based on aluminum. Alloys of copper base and those of magnesium base are growing in importance, but production is relatively limited. Alloys of lead and tin are readily die-cast but are of small commercial importance. Ferrous alloys are not die-cast commercially. In Table II are listed properties of these and of other alloys. Not including plastic rods and tubes, there is a far wider choice of materials avail-

Parts die-cast in alloys based on zinc, aluminum or copper. Each piece could be duplicated in shape, though probably not with equal economy, on the screw machine. For some of these parts supplementary operations would be required

able for screw machine products than for die-castings.

WEIGHT. Relative specific gravities of materials used for die-castings and screw machine products are given in Table I. Since the wrought materials for screw machine products are stronger than those for die-castings and the gravities are about the same, it follows that for a given strength a lighter screw machine product can be made, especially in steel. But, if sections must be the same, the die-cast part will be somewhat lighter than a steel or brass screw machine product. It is possible to further lighten the screw machine product if full advantage be taken of the greater strength attainable through heat-treatment. The weight of metal required to make the part is likely to be greater for the screw machine product than for the die-casting, because of greater scrap loss.

STRENGTH AND DUCTILITY. Most cast materials are inferior to the corresponding wrought non-ferrous materials in the significant physical properties and die-castings in general are not an exception to this rule. The commoner die-casting alloys are inferior to steel in strength and hardness, and have lower softening and melting points. All die-cast forms are subject to some porosity the location of which generally can be controlled so as not to come in highly stressed sections. Porosity is minimized and often so nearly eliminated by good casting technique, that for a wide range of applications it can be disregarded. Naturally, wrought materials are free from porosity. If the die-casting, which is always non-ferrous, be compared with the steels most commonly used for screw machine products, the difference in physical properties is still greater. It will be observed from Table II, however, that brass die-castings can be had with a tensile strength greater than that of mild steel. It may be noted, also, that the differences as between wrought and die-cast non-ferrous alloys in tensile strength is not very wide when the former are in the soft annealed condition which is preferred for production on the screw machine.

It is also significant that the zinc alloys and those in brass are remarkably high in impact strength and quite high in ductility for cast materials. It is thus feasible to perform such operations as punching and spinning on zinc alloy and on certain other die-castings, much as can be done on wrought metals. In both classes of product, however, a considerable proportion of the total output is for types of applications in which the strength and ductility of either die-cast or wrought materials is ample to

Table I—Design Factors Compared for Die-Casting and Screw Machine Products

	DIE-CASTINGS	*SCREW MACHINE PRODUCTS*
MATERIALS	Zinc alloys lowest in cost and most widely used. Aluminum alloys next in cost and extent of use. Alloys of copper, magnesium, tin and lead used for some purposes — No ferrous materials available	Free-machining steels lowest in cost and most widely used. Many other steels available. Brass widely used and many other non-ferrous alloys are available. Some alloys suited for die-casting are not available in rods or tubes. Some plastics used
WEIGHT (*Specific gravity*)	Zinc alloys average 6.7 Aluminum alloys average 2.75 Magnesium alloys average 1.81 Copper alloys average 8.40 Lead alloys average 10.65	Free machining steels about 7.86 Free cutting leaded brass 8.50 Range of other materials about the same as for corresponding die-casting alloys
PHYSICAL SHAPES	Generally comparable to screw machine products. Often include detail differences such as ribs, bosses, irregular recesses, tapered flutes and serrations not capable of duplication by screw machine. Readily made hollow at both ends without extra operations	Primarily generated by rotation and those having section of stock used, but secondary operations often performed by attachments permit certain variations from rotational shapes. Shaping at cut-off end limited except in secondary operations. Can have interior and exterior undercuts not feasible on die-castings
SURFACE SMOOTHNESS	Often cast so smooth that only buffing is required before plating but average may be inferior to that on screw machine products	Depends on tools used and other factors but is often remarkably good and can be brightly burnished. Surfaces normally brighter and often smoother than for the castings
DIMENSIONAL ACCURACY	Usual commercial tolerance ± 0.001 to 0.003 in. per in. on most important dimensions	Usual commercial tolerances ± 0.001 to 0.003 in. on most significant dimensions regardless of size of piece
TOOLING COST	Casting die and tools for flash removal likely to exceed tooling cost on screw machine. Tool upkeep usually low or negligible	Often an important consideration in total cost. Tool upkeep often rather high, but total tooling costs are likely to be lower than for die-casting
PRODUCTION RATE	Usually 50 to 1,000 cycles an hr. and one to several castings per cycle with one man usually in continuous attendance	Usually 30 to 3,600 pieces produced per hr., depending on number and character of operations, size, and material used. One man usually attends several machines
LABOR COST	Likely to be higher for die-casting because of extra operations needed subsequent to casting and use of non-automatic machines	Likely to be lower as machines are automatic and often complete all machining required
MATERIAL COST	About 10¢ per lb. for zinc alloys About 12 to 17¢ per lb. for alum. alloys About 10 to 14¢ per lb. for copper alloys usually lower than for screw machine products except when latter are in low cost steel	About 4.5¢ up per lb. for free cutting low carbon steels About 15¢ up per lb. for free cutting brass usually exceeds that for die-casting unless low cost steel can be used
WASTE IN SCRAP	Slight because scrap is remelted as are also any rejects produced	Often considerable especially if much metal is removed, but non-ferrous scrap usually has good resale value
MACHINING COSTS	Varies from slight to considerable depending on requirements, all in secondary operations	Usually moderate to low, but a part of basic production. Secondary operations, frequently needed, costs depending on requirements
COST OF APPLIED FINISH	Usually about the same as for screw machine products but more polishing and buffing may be needed for plating	Usually about the same as for die-casting but skiving or burnishing often save in polishing and buffing of parts to be plated
STRENGTH AND DUCTILITY	Nearly always inferior to steel but for corresponding non-ferrous alloys the differences are often slight though generally favoring the wrought materials used in screw machine products	Much superior for steel and usually somewhat higher for those non-ferrous alloys which correspond in a general way to certain die-casting alloys
HARDNESS AND CREEP	Zinc and aluminum alloys range from 60 to 83 Brinell; brass 120 to 180. Zinc alloys subject to creep at atmospheric temperatures	Steel 130 to 241 or higher Brinell. Copper alloys (brass, etc.) — 10 to + 105 Rockwell "B" (1/16 in. ball 100 kg. load) Cold flow seldom significant

meet requirements, since the parts are not highly stressed. When this is true, of course, the designer is likely to be more concerned with cost and other considerations than with mechanical properties.

There are available for screw machine products materials which are more resistant to corrosion than any available for die-casting, although both types include materials ranking high in this regard. Most ferrous materials are subject to red rust, whereas die-castings are not.

In general, the die-casting cannot be heat-treated with benefit in physical properties because it is subject to blistering at ordinary heat-treating temperatures. Many screw machine steels can be heat-treated with important improvements in strength and hardness.

HARDNESS AND CREEP. When hardness is desired, steel has a marked advantage over the non-ferrous alloys used for die-casting, and also those in wrought form used for screw machine products. Hardness, however, affects machining properties and is sometimes a disadvantage; Table II gives some relative values. Zinc alloy die-castings and those of lower melting point are subject to slight creep, that is, cold flow, even at normal atmospheric temperatures. The effect of creep is seldom a factor in design unless rather high stresses are imposed, especially in bending and tension. Creep is said to be negligible or nil in die-cast alloys of higher melting point and the same is probably true of all important wrought materials used for screw machine products, as far as ordinary temperatures are concerned.

PHYSICAL SHAPES. There is a greater limitation as to shapes readily produced in the screw machine product than in die-casting. Die-cast products are by no means confined to shapes involving surfaces of rotation; variations in shape are almost unlimited. Parts can be die-cast with either internal or external threads, although it is often more economical to tap than to cast internal threads. Cast external threads usually require chasing, especially when the die is parted in a plane through the axis of the threaded portion. Die-castings are cast close to size, hence when machining is necessary only light cuts are required. Flats, slots, cross holes, specially shaped contours, gear teeth, keyways, splines and various projections which require from one to several extra operations in screw machine products are readily made in the die-casting.

Certain odd shapes or contours can be produced in screw machine products, of course, by using rods or tubes drawn or extruded in special sections, as with flutes or gear teeth. Squares, hexes and other standard shapes of bars are often employed to avoid cutting flats. However, the shaping or special contour must come at the maximum diameter of the piece; or if tubes with irregular holes be used, the shaping is at this hole and not at a larger diameter. These limitations do not apply to the die-casting.

Die-castings can often combine in one piece features which would require two or more parts made partly or wholly on a screw machine. Holes of almost any shape whether axial or not, are readily cored in the die-casting; whereas holes in the screw machine product, unless formed in secondary operations, usually must be axial and circular.

Screw machine products do not require draft, such as is commonly needed on a die-casting, and can have undercuts in holes. Knurling of complete circumferences is not usually feasible on the die-casting unless it be done in a supplementary operation after casting or unless a straight knurl can be used with serrations parallel to the motion of the die part which forms them.

[*Editor's Note*—Part II of this article, which will be presented in the July number, will discuss other factors that have a bearing upon choice as between die-castings and screw machine products, such as dimensional accuracy, surface smoothness, appearance, production rates and costs.]

Table II—Properties of Materials Suitable for Die-Casting and for Screw Machine Products

Type of Material	Tensile Strength In 1,000 lb. per sq. in.	Yield Point In 1,000 lb. per sq. in.	Elongation % in 2 in.	Reduction of Area Per Cent	Hardness Brinell Number	Machine-ability
DIE-CASTING ALLOYS						
Zinc Alloys	40–48	.....	3.0–5.0	.....	74–83	Excellent
Aluminum Alloys	30–36	.....	1.0–2.0	.....	60–80	Fair to Good
Magnesium Alloys	30–33	17–22	1.0–3.0	.....	60–62	Excellent
Copper Alloys (Brass)	55–95	35–72	8–20	.....	120–180	Poor to Good
WROUGHT NON-FERROUS ALLOYS*						
Leaded Brasses	38–55	16–27	20–60	.....	.....	Excellent
Other Brasses and Bronzes	54–55	24–70	25–50	.....	.....	Fair to Excellent
Pure Copper	32–35	10	45	.....	.....	Poor
Aluminum	13**	5**	45	.....	23	Excellent
COLD DRAWN ANNEALED STEELS						
Free Cutting Steels	70–120	60–100	10–25	30–55	140–235	Good
Carbon Steels	70–120	60–100	10–25	30–55	149–235	Fair
Nickel Steels	80–125	60–110	10–25	35–60	170–235	Fair
Nickel Chromium Steels	85–110	70–100	12–20	40–60	179–228	Fair
Molybdenum Steels	90–115	80–100	15–25	40–60	183–228	Fair
Chromium Steels	90–115	70–100	12–25	35–60	170–241	Poor to Fair
Chrome-Vanadium Steels	95–115	80–105	12–20	35–60	179–228	Fair
Stainless Steels	65–90	35–55	25–60	55–70	130–185	Fair

* Soft grades. Hard grades have higher physical properties.

** Commercially pure aluminum in soft grade. Harder grades are stronger and machine freely. Wrought aluminum alloys in heat-treated form having tensile strengths up to 60,000 lb. per sq. in. or above are available and some such are even freer machining than the commercially pure metal.

Cabinet for deWald radio and assembled housing for the Taylor Instrument Baroguide, both cast in Catalin

CAST PHENOLICS

Design Data and Fabrication Cost Analysis

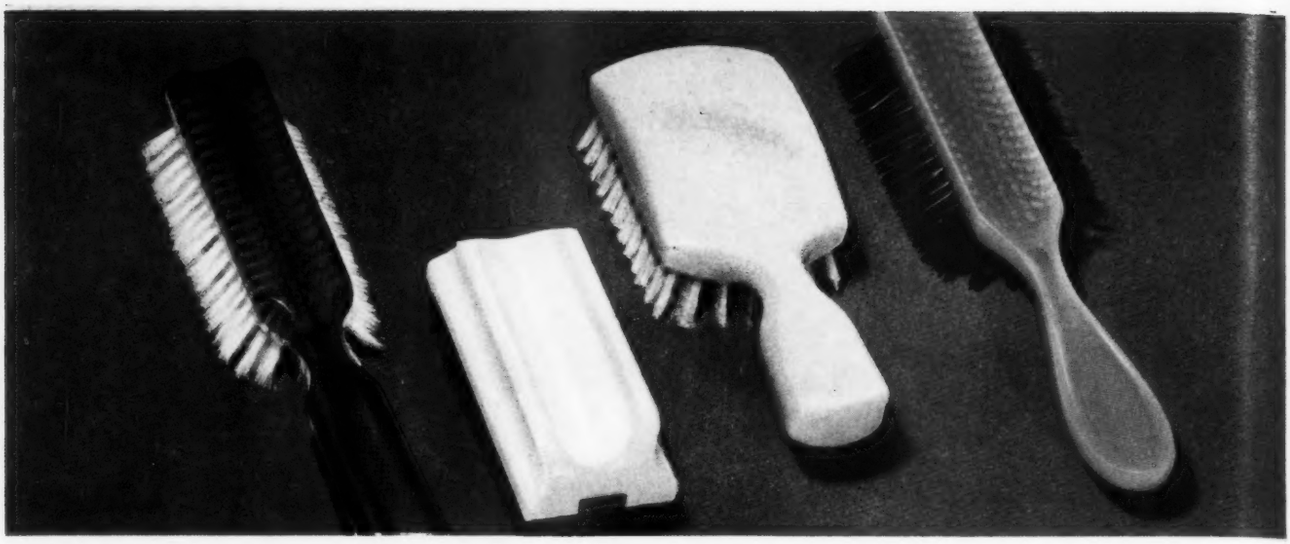
IN PREPARING to design a product consisting wholly or in part of plastic material, the engineer must investigate four important points. (1) The physical and chemical properties of the plastic—will it withstand the conditions to which it will be subjected, and for how long. (2) The design limitations of the plastic material, which relate somewhat to its physical properties but which principally depend on production methods. (3) The economy of production, which takes into consideration the cost of tooling up, the time involved in tooling, and the quantity of parts to be obtained from the initial tooling. (4) The cost of the plastic product in relation to the market of similar products made of other materials, considering what advantages the plastic product may have from the points of structural qualities, and appearance.

In many instances, it may be possible to sacrifice one or two strength considerations for appearance. From the standpoint of appearance, cast phenolic plastics are particularly important.

Since no filler is used in the production of phenolic resins, they have great depth of color and richness of luster. As a further corollary, cast resins have good machining qualities thus permitting considerable flexibility of design and low initial set-up cost.

When evaluating casting with compression and injection molding, the following factors are important. Initial tool costs on cast phenolic are considerably lower than on molding. Thus cast phenolic will often be economical on runs which compared to the average compression mold run, are relatively short—several thousand or less. This factor is important in the design of large parts such as pilasters, since as the size of a given part goes up, the cost of compression molds increases almost geometrically. On the other hand, cast phenolic parts may require more fabrication than molded parts, which generally only require removal of flash and a light buffing. A point in favor of cast phenolic is that standard forms are available for many parts, and thus initial tool cost can be eliminated.

Unlike other plastic materials, cast



Cast phenolic, particularly attractive for translucent effects, as used for brush handles—materials are Catalin and Opalon

Properties of Available Cast Phenolics

	General Purpose Opaque and Translucent	General Purpose Transparent	Special Purpose Chemical Resistant	
			Transparent	Opaque
Specific Gravity	1.30	1.30	1.27	1.70
Tensile Strength, lb. per sq. in.	3,500–5,000	3,500–5,000	5,000–7,000	6,000–9,000
Flexural Strength, lb. per sq. in.	7,000–9,000	3,000–7,000	9,000–14,000	10,500–12,000
Impact Strength. Energy to break in ft. lb. ½ in. sq. rod. Notched.	0.15–0.20	0.175–0.225	0.15–0.20	0.19–0.24
Heat Distortion deg. C.	40–45	40–45	55–65	75–76
Water Absorption in 48 hr. percent Room Temperature	0.5–2.0	0.015–0.60	0.4–0.6	0.4–0.6
Brinell Hardness, 5mm. Ball	20–30	15–20	35–40	35–40
Dielectric Strength. Step.—0.125 in. thick. Volts per mil	200–250	175–200	275–300	120–142
Power Factor 60 cycles	0.25–0.35	0.25–0.50	0.10–0.15	0.07–0.074
Dielectric Constant 60 cycles	15–20	20–30	7–8	30–31
Loss Factor 60 cycles	2.5–8.0	10–16	0.70–1.0	2.07–2.20

Water Absorption after 8 days immersion—0.0057–0.0081 grams per sq. in. of surface.
Water Absorption after 80 days immersion—0.0132–0.02 grams per sq. in. of surface.
Softening Ranges—176 deg. F. to 212 deg. F.
Effect of Dry Heat—withstands up to 160 deg. F. with slight hardening and shrinkage.
Effect of Moist Heat—withstands up to 160 deg. F. with slight hardening and shrinkage.
Effect of Ageing (Indoors)—slight increase in hardness, improves in electrical properties.
Effect of Ageing (Outdoors in sunlight, one year)—slight increase in hardness, yellowing and fading of non-fast colors. Mechanical and electrical properties not appreciably changed.
Effect of Water (Cold)—none.
Effect of Water (Hot)—withstands intermittent washing with hot water. Slight deterioration after 100 hr. complete immersion.
Effect of Strong Acids—slowly attacked.
Effect of Alkali (Weak)—can be washed with soap and soap powders. Slowly softens when immersed in alkaline solutions, showing slight discoloration.
Effect of Alkalis (Strong)—slowly attacked.
Effect of Alcohols—insoluble—slight swelling in ethyl alcohol after long immersion.
Effect of Mineral Animal or Vegetable Oils—none.
Effect of Hot Flames—chars but does not support combustion.
Solubility—insoluble in water, organic solvents and acids. Slightly soluble in alkalis.
Tendency to Cold Flow—none.
Opacity Range—opaque to transparent, all colors.
Colors—innumerable.
Forming Qualifies—good.
Machining Qualities—excellent.
Light Transmission (⅛ in.—40 watt source)—70 to 92 percent.
U. V. Transmission (2800–3200A)—18.3 percent.
Index of Refraction—1.46.

phenolic resins are not supplied in powder form and cannot be injection or compression molded to final form. Phenol and formaldehyde combined with a catalyst are processed in huge kettles under a vacuum. The basic ingredients in liquid form are cooked starting a process of polymerization which continues throughout successive production steps to the finished casting.

In the next production step, the mixture is subjected to a vacuum, and the water which is a product of the phenol-formaldehyde reaction, is drawn off. After the proper dyes are added to give color, the liquids are drawn from the bottom of the kettles and poured into lead molds. The molds are then placed in vulcanizing or curing ovens where polymerization continues until fully cured castings having a Rockwell hardness of approximately plus B 50 to 100 are produced.

The castings are removed from the molds by air hammers. The temperature of the ovens is maintained at 80 deg. C. throughout the polymerization,

and no pressure is involved in the molding at any point. However, the casting process is particularly interesting to the designer inasmuch as it sets the limitations within which he must work and affects production cost.

Cast Phenolic Molds

Three types of molds are employed in the production of phenol formaldehyde cast resin material: straight draw molds; split molds; cored molds.

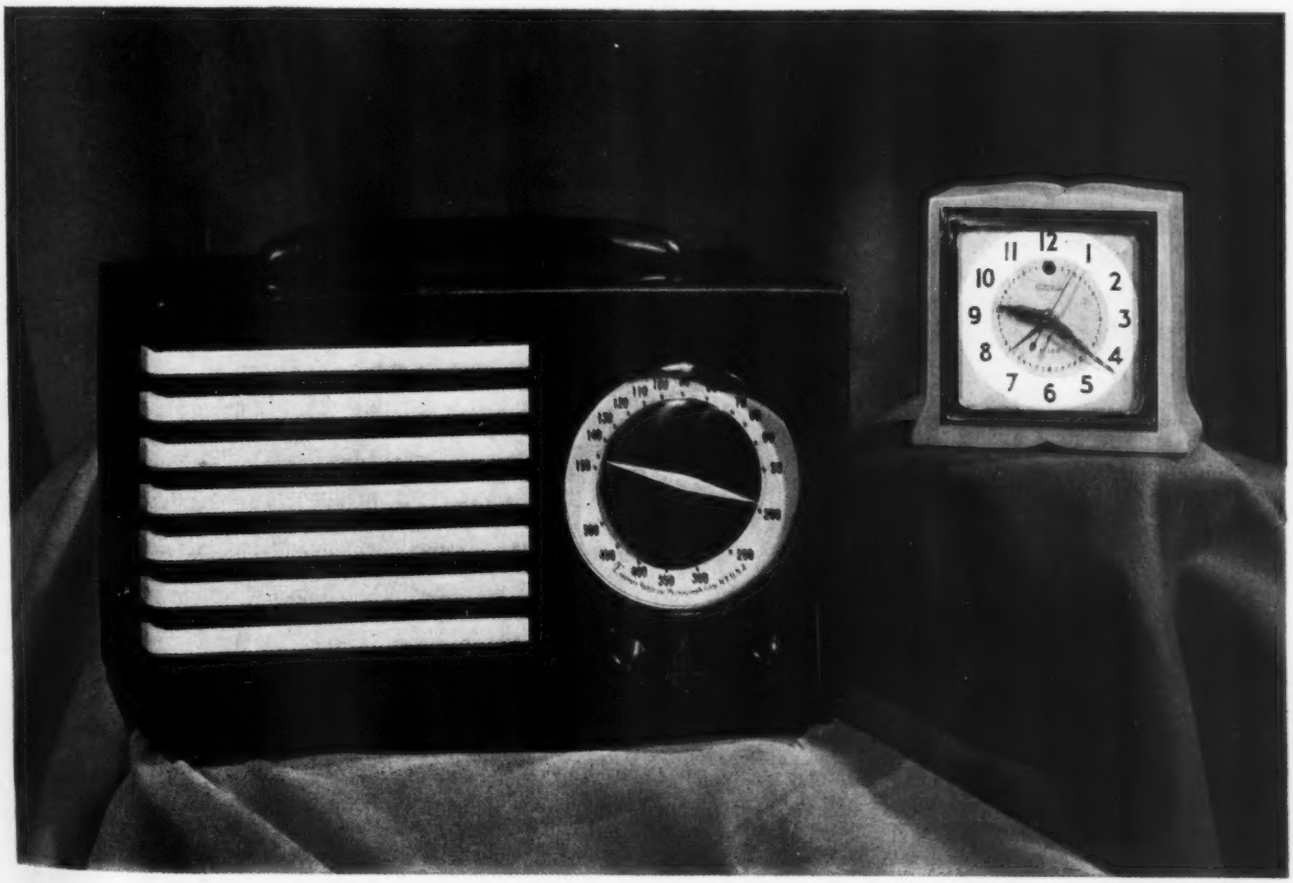
Draw Molds. Approximately 90 percent of the phenolic castings produced are cured in straight draw molds. In this method, a steel dipping arbor or mandrel is machined to the specifications of the finished casting and attached to a plate and handle. The mandrel is then dipped into a pot of molten lead which flows up the sides of the steel and which cools on removal from the pot. When stripped from the arbor, the open lead mold is ready for use. It is obvious that no under-cuts can be designed into a casting of this type. Flutes, scallops, beads and other design accessories may be incorporated in the castings, but they must run in the direction in which the castings are removed from the mold. Tubes, boxes, round rods and rods cabinets, pilasters and other large parts having profile shapes of all types may be produced in this manner. A taper of approximately 0.0015 in. per in. is necessary in order to break the vacuum, reduce friction, and permit the casting to be removed from the mold.

Because of this taper and because the castings are forcibly removed from the molds, there are certain minimum requirements for wall thickness. For large radio cabinets, box castings, long tubes, plate castings and similar items, the walls of the castings must be $\frac{3}{16}$ in. thick. For small tubes, small boxes and other articles which can readily be removed from the molds, a 5/32 in. wall is required. In the case of small castings, such as hollow bottle closures, $\frac{1}{8}$ in. walls will suffice. The taper in all cases is plus from these dimensions.

Certain factors should be borne in mind when designing cast phenolic parts. Where large flat surfaces are required, it is desirable to use mottled or marble-like colors, rejects will be reduced since minor imperfections in color will not usually be discernible. Corners or edges should be given a slight radius, not only to facilitate the casting operation, but as a help in reducing machining time after the part has been withdrawn from the mold.

Cored Molds. To obtain half-spherical hollow castings, pilasters with compound curves, and large castings with decorative faces, cored molds are used. In this case a dipping arbor is made to the outside specifications of the finished piece: if a half-spherical hollow casting is specified, the arbor and consequently the lead molds will have the dimensions of the periphery of the sphere. Into this mold, and properly centered $\frac{3}{16}$ in. to $\frac{1}{4}$ in. away from the lead, is placed a metal core, the outside dimensions of which are identical with the inside dimensions required of the hollow sphere. Between the core and the lead mold liquid resin is poured for curing in the ovens. When the material reaches the proper hardness, the core is removed and the casting drawn from the lead mold.

Although this production method permits a considerable latitude in designing, it is not recommended for units smaller than 4x4x4 in. The amount of handling per casting remains the same and the number of cores required remains the same, but the material over



Emerson radio having mottled walnut Opalon cabinet and electric clock housing cast of coral-colored Opalon

Radio cabinet of Bakelite cast resin; compression-molded urea grille is attached with vibration-proof fastenings; bezel and pointer are cellulose acetate

which these costs are amortized drops to a quantity so low that the price per pound is prohibitive. No production quantity limitations are placed on this type of molding, and the cores may be used for an indefinitely large production. No taper is required on castings made in this manner, and the walls can be of any thickness above $\frac{3}{16}$ inch.

An important feature in this connection is that walls can be made as thick as is consistent with the structural requirements of the product without fear of under-curing, defective castings, or distortion. However, there are virtually no limits to how large a casting may be; units 20 in. square and 12 in. deep are now included in regular production. Pilasters as long as 36 in. and having a periphery of 18 in. are likewise standard production.

The shrinkage factor is particularly important in designing castings of large size since it amounts to approximately 1 percent and can have a serious effect on the assembly unless the phenolic units are properly mounted and assembled.

Where extremely long pieces, such as pilasters, are involved, the castings in final assembly should be suspended and fastened at either the top or the bottom, riding in grooves along the sides so that expansion and contraction with temperature changes and any shrinkage will not set up stresses which result in cracking.

It will be noted from the description of the cored molding process that it concerns open molds, i.e., the material is poured into the openings between the core and the lead mold.

Split Molds. In split-molding, the process involves two-piece molds closed at all points except for the gate into which the liquid resin is poured. Undercut designs may be incorporated into articles produced by the split-mold method so long as the undercuts are not more than two-directional. It must be possible on opening the mold to draw each half away from the casting without incorporating under-cuts in the respective half.

In this case die-casting machines are used for production of the lead molds, the machines making one-half of the mold at a time, each of which interlocks with the opposite half of the neighboring mold.

The successive sections are clamped together, liquid resin is poured into the gates, and the material is cured in the usual manner. Walls cannot be less than $\frac{3}{16}$ in. thick. To economically produce the item there should not be more than ten to the pound.

From the foregoing it will be seen that expensive tools are not necessary in the production of cast phenolic plastics. Dipping arbors vary from $90 to $375, depending on the size and intricacy of the design. Arbors and tools for producing cored castings vary in cost from $275 to $800, and tools for producing split-mold castings range between $250 and $550.

On these rough castings, varying amounts of fabrication are necessary. From the split-mold castings the gate and the fin must be removed. From the cored mold and straight draw mold castings the flash which varies from $\frac{1}{8}$ to $\frac{3}{16}$ in. at the open end of the mold must be machined off.

Because of the machining qualities of cast phenolics, these operations can be performed with a minimum amount of expense. The flash is removed either with a bandsaw, an abrasive cut-off wheel, or a dry or wet grinding disk, depending on the application for which the casting is intended. It is then drilled and tapped for assembly or drilled for stapling, then routed, beveled, shaped, polished, and otherwise handled much the same as wood or metal, using substantially the same tools and equipment.

Several methods of fastening the castings are available. Liquid phenolic cements are used in some assemblies, but machine screws, drive screws and self-tapping screws can also be used. It is not recommended that phenolic castings or sheets be joined to material having a different coefficient of expansion and contraction. For example, a piece of phenolic sheet stock joined to plywood will crack should the plywood swell with the absorption of moisture.

Colors available vary from water clear to jet black, in a wide variety of stock shapes, including round, square, octagonal and hexagonal rods, cylinders, sheets and plates. Sheet material is available in any size up to 24x48 in. and in any thickness above $\frac{1}{8}$ in. It can usually be obtained either polished or unpolished.

An important feature of cast phenolic plastics is translucency, characteristic of practically all colors except the dark browns, blues, and blacks. Advantage is taken of this translucency in decorative pieces for fluorescent lighting fixtures, coin-operated phonographs, china housings and in interior lighting effects.

[The editors acknowledge the extensive cooperation of Catalin Corporation and the valuable suggestions submitted by Monsanto Chemical Company and the Bakelite Corporation.]

SHIMS FOR BEARINGS

Selection and Applications of Various Types

ALBERT B. WILLI
Chief Engineer, Federal-Mogul Corporation

IN MANY MECHANISMS shims are a necessity, since they provide quick and inexpensive means for adjustment of clearances. Thus, with split-sleeve bearings, the use of shims in assembly permits later adjustment for bearing and shaft wear, and also allows the user to adjust oil clearances as required by service conditions. Many engineers use shims in assemblies where non-aligning elements must be coupled, or for adjustment of ball bearings, roller bearings, and bevel gears. Outstanding examples wherein the use of shims speeds up production, through reduction of precision machining operations and reduction of assembly time, may be found on pages 197-199 of April PRODUCT ENGINEERING.

The simplest sleeve bearing adjustment shim is of flat metal, stamped to desired shape and having a thickness of either 0.002 or 0.003 in. The material is usually brass of the composition 65 percent copper, 35 percent zinc; ½ to ¾ hard. Standard thickness tolerance on loose-leaf brass shims of 0.002 and 0.003 in. thickness is plus or minus 0.0002 in. Steel shims are also available in special S.A.E. 1010 cold rolled stock; standard thickness tolerance on steel shims is 0.00015 in. plus or minus.

Loose-leaf shims, generally used in packs of two or more, are somewhat difficult to work with, since the correct number must be counted out for each assembly and the stack is difficult to keep in matched alignment during assembly operations. The hazard of dirt adhesion to the individual leaves is always present. With laminated shims having two or more laminations or leaves bonded together in such a manner that individual leaves can be readily peeled off, these difficulties are eliminated. Smooth and flat surfaces remain.

In the composite construction shown in Fig. 1, three basic leaf members are used. The upper member is a unit pack of brass laminations 0.002 or 0.003 in. thick, bonded together. The other two members may be either brass or steel, or one of each. An adjustment feature is provided by the laminations since one or more can readily be peeled off as required. The thickness of this member cannot be less than 0.020 in., al-

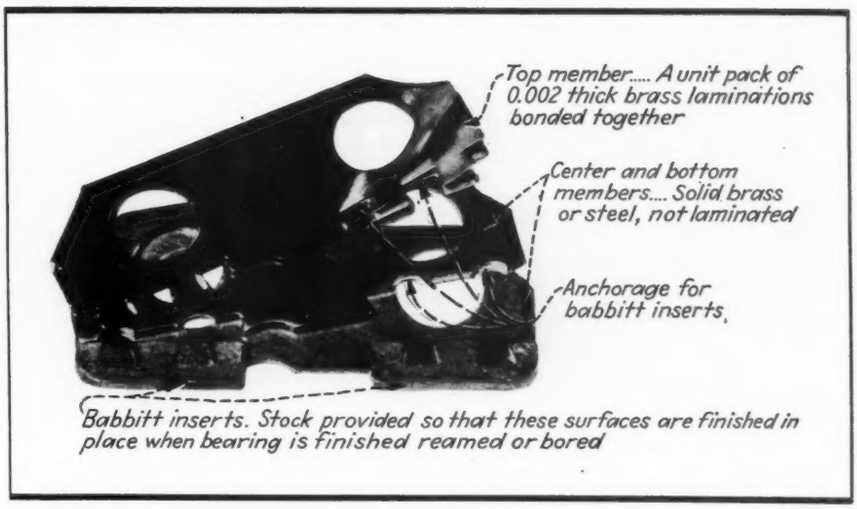

Top member..... A unit pack of 0.002 thick brass laminations bonded together
Center and bottom members..... Solid brass or steel, not laminated
Anchorage for babbitt inserts.
Babbitt inserts. Stock provided so that these surfaces are finished in place when bearing is finished reamed or bored

Fig. 1—Composite shim construction with babbitt insert facings at shaft contact surface (Laminated Shim Company)

Table I—Commercial Tolerances for Laminated Shims

(Courtesy Laminated Shim Company)

NOMINAL TOTAL THICKNESS-IN.	NO. AND SIZE OF INDIVIDUAL LEAVES	TOLERANCE ON TOTAL THICKNESS-IN.
0.004	2—0.002 in. thick	±0.00025
0.006	2—0.003 in. thick	±0.00025
0.006	3—0.002 in. thick	+0.00025/−0.0005
0.009	3—0.003 in. thick	+0.00025/−0.0005
0.008	4—0.002 in. thick	±0.0006
0.012	4—0.003 in. thick	±0.0006
0.030	15—0.002 in. thick	±0.002
0.030	10—0.003 in. thick	±0.002

Table II—Standard Tolerances on Total Thickness

(Recommended Thickness of the Laminated Member)

SHIM THICKNESS IN. (NOMINAL)	STANDARD THICKNESS—IN. TOLERANCE	THICKNESS OF LAMINATED LEAF MEMBERS—IN.
0.062	±0.005 These tolerances	0.020 Individual lamina-
0.093	−0.000 can be reduced	0.032 tions may be either
0.125	−0.000 but at increased	0.032 0.002 or 0.003 in.
0.187	±0.005 cost	0.047 thick.

NOTE: Shims of this construction can be built to a minimum thickness of 0.062 in. although 0.093 in. is preferred — as a minimum. The usual thicknesses are 0.062 in., 0.093 in., 0.125 in. and 0.187 in., but thicknesses up to 0.500 in. are possible.

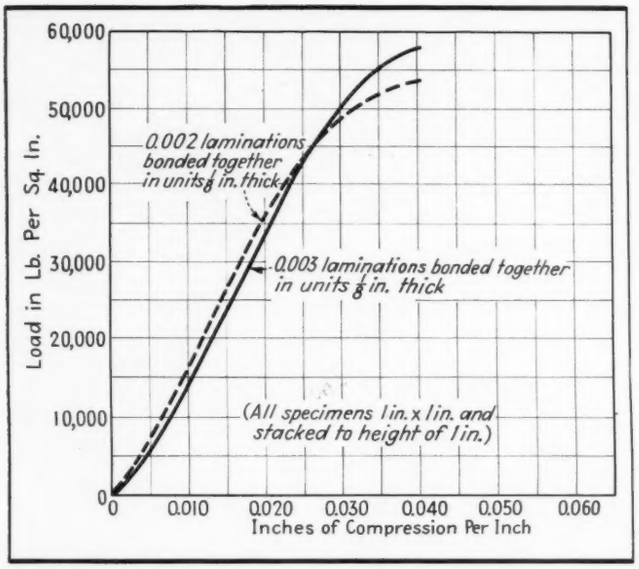


Fig. 2—Compression of laminated brass shims, as reported by the Laminated Shim Company

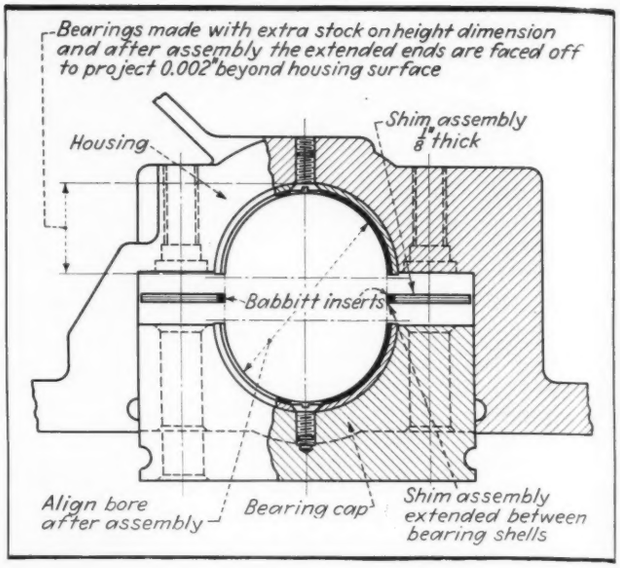

Bearings made with extra stock on height dimension and after assembly the extended ends are faced off to project 0.002" beyond housing surface
Housing
Shim assembly 1/8" thick
Babbitt inserts
Align bore after assembly
Bearing cap
Shim assembly extended between bearing shells

Fig. 3—Shim assembly, 1/8 in. thick, provides clearance adjustment in this typical application to split-sleeve bearings

though 0.032 in. is preferred to facilitate manufacture. The three leaf members are locked together through babbitt inserts, an efficient and patented system of anchorages. Shims of this construction can be built to a minimum thickness of 0.062 in., although 0.093 in. is preferred as a minimum.

With all-brass or all-steel shims having a total thickness of 1/16 in. or more, the space between the shaft and the edge of the shim is likely to be so large that an excessive amount of oil may be lost from the bearing by leakage. When this type of shim is brought close to the shaft, so as to reduce this leakage, there is danger that the shaft will rub the shim surface and become scored and ridged. The babbitt inserts should be made with a stock allowance for finishing off when the bearing is bored or reamed to size at assembly. The clearance between shaft and shim is thus the same as the bearing clearance and there is no danger of shaft damage.

With this type of shim, there is little danger of harmfully reducing bearing oil clearance caused by shim compression when the cap bolts are tightened. The laminated leaf member would be most likely to compress under load, because of the bonding medium between the individual laminations. However, tests reported by the Laminated Shim Company, see Fig. 2, show that the compression is so small that it can be disregarded.

A typical well-designed application of split-sleeve bearings in combination with shims for oil clearance adjustment is shown in Fig. 3. The fundamentals of this design, which may be made a general recommendation, are: bearings are align bored (or reamed) to size in place; bearings are made with extra stock on the height dimensions and after assembly the extended ends are faced off or fitted to project 0.002 in. beyond the housing and cap surfaces. The amount of stock allowed for fitting depends on the uniformity of the depth of the half bore in the housing and in the cap, and the method to be used in fitting, whether hand filing or milling. The 0.002 in. projection insures that the bearings will be tightly seated when the bolts are drawn down.

The 1/8 in. thick shim assembly follows the general construction shown in Fig. 4, and is built up with two leaves of 3/64 in. thick brass, and one 0.032 in. unit pack made up of 0.002 in. brass laminations. Babbitt inserts are provided at the shaft contact surface. As a further advantage the babbitt inserts are finished off when the bearings are bored, thus controlling oil leakage and affording protection to the shaft. Oil clearance can be reduced by peeling off one of the 0.002 in. thick laminations, or if additional clearance is required laminations can be added.

The success of an assembly in which bearings are used with extended shims and in which the height fitting operation on the bearing parting faces is omitted, is problematical. It is easy to manufacture bearings to close height tolerances, but it is not easy to maintain uniform heights of the half-bores in housings, rods or caps as at *A* and *B* Fig. 5, since the half-bore height *B* may be, and frequently is, as much as 0.015 in. greater than the complementary half-bore *A*. As a consequence, several harmful conditions will result. The upper bearing shell 1 will not be firmly seated, since a gap will exist at *C* between the parting faces of the bearings and the shim surface. Also, the lower shell 2 will extend an excess amount

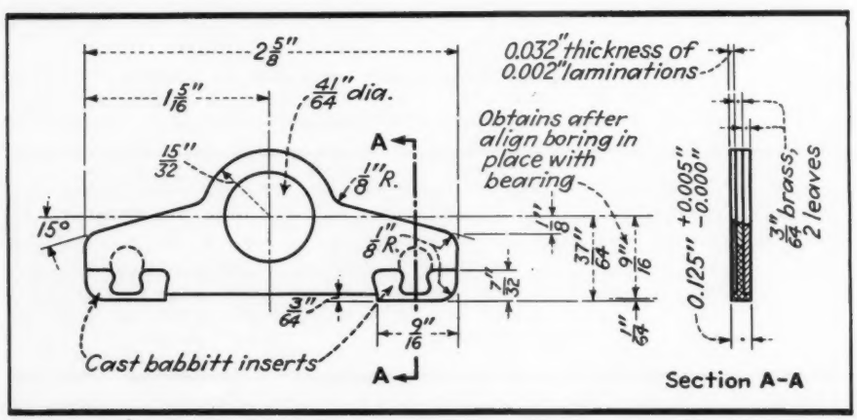


Fig. 4—Composite babbitt-faced shim assembly (Laminated Shim Company)

into the shim space at *D* and the shim will be distorted and perhaps bent to the extent that the bearing will not be able to seat properly adjacent to the shim surfaces.

When good back contact is not provided for a bearing, heat flow away from the rubbed surface is seriously impeded or completely stopped off. An overheated bearing is a guarantee of short life.

If housing and cap half-bore heights can be maintained uniform within 0.0005 to 0.001 in., it is possible to eliminate the fitting operation on the parting faces in a bearing assembled with shims.

The combination of precision insert bearings and a pack of thin shims—say 0.006 to 0.009 in. thick, does not seem practical, since many precision insert bearings have comparatively light walls, and thin shims which extend between the shells are likely to have the extended portions pinched off when the bolts are tightened.

Of course, with precision insert bearings, it is not necessary to maintain extremely close uniformity in the half-bore heights of housing and cap. Because of the difficulties in maintaining uniformity in these dimensions, the same objections prevail with respect to the use of extended shims with precision insert bearings.

With light-wall steel-backed precision-insert bearings, thin shims which extend only to the outside diameter of the bearing, Fig. 6, have been rather extensively used. If the rod or housing bore are finished to size with the final shims in place, the assembled bearings will have an inside diameter of the roundness and size intended and the oil clearance will be correct. If, however, the rod or housing bores are sized with a "process" or "dummy" shim in place and the production shim is installed at assembly, the tolerances on shim thickness become a problem which affects the oil clearance. The bearing shown in Fig. 6 is intended to be used with a minimum of 0.0005 in. oil clearance. If three loose-leaf shims are used, the possible total thickness variation will be plus or minus 0.0006 in. If a unit pack of laminations is used thickness variation will be 0.00075 in.

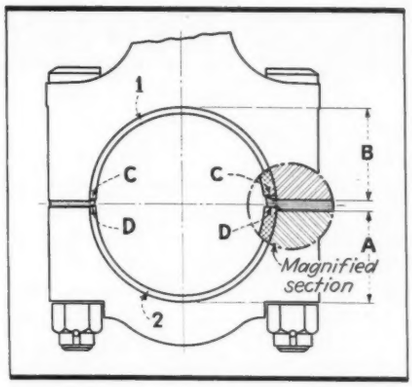


Fig. 5—Bearings without stock on parting faces are not satisfactory with shims

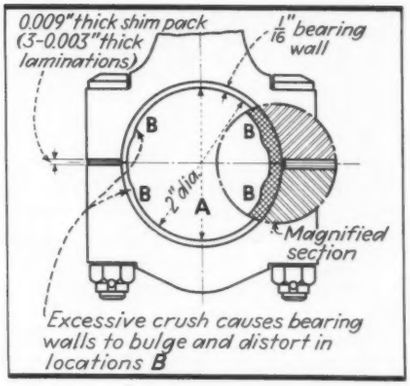


Fig. 6—Connecting rod bearing with thin shim packs which extend only to rear

In a precision insert bearing installation, the oil clearance is a function of the tolerances on housing bore, bearing wall thickness and shaft diameter. When shims are introduced, their thickness variation increases the possible spread between minimum and maximum oil clearance dimensions and usually requires checking every bearing for size in a pre-assembly operation and necessitates adjustments in the thickness of the shim pack, thus defeating the purpose of the precision insert bearing.

When compensating for wear by removing a shim, another possible difficulty may arise. The height *A*, Fig. 6, of a pair of bearings is made slightly greater than the housing bore. If this height is too great, the bearing wall bulges into the clearance space when

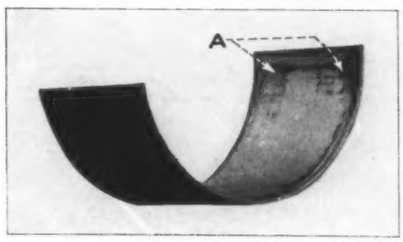


Fig. 7—Excessive crush caused bearing wall distortion, shown by dark area A. Shaft was in contact with bearing in this area only

the bolts are tightened at points *B* Fig. 6, creating high-pressure areas the full length of the bearing. A condition of this sort is shown in Fig. 7. Since the total load is carried on a small area of the bearing, premature failure will occur in area *A*.

When a shim is removed, bearing "crush" is increased. If the reduction in the bearing dimension vertically is greater than the amount of bulge, probably no trouble will result. Conversely, if the bulge at any point adjacent to the parting is greater, then premature failure within the area affected can be expected.

In the construction shown in Fig. 8, a precision insert bearing is used in a connecting rod originally built to use extended shims. The bearings are full halves. The original shim space in the rod is taken up by solid steel spacers which extend only to the back surface of the bearing. Although the adjustment feature is eliminated, this construction makes possible emergency replacement of bearings with a minimum of delay.

Extended shims also serve the useful function of locking bearings against rotative movement. Sometimes the force applied is so great that the parting faces of the bearing will imbed themselves into the surface of an all-brass assembly, causing harmful distortion and deformation of the shim.

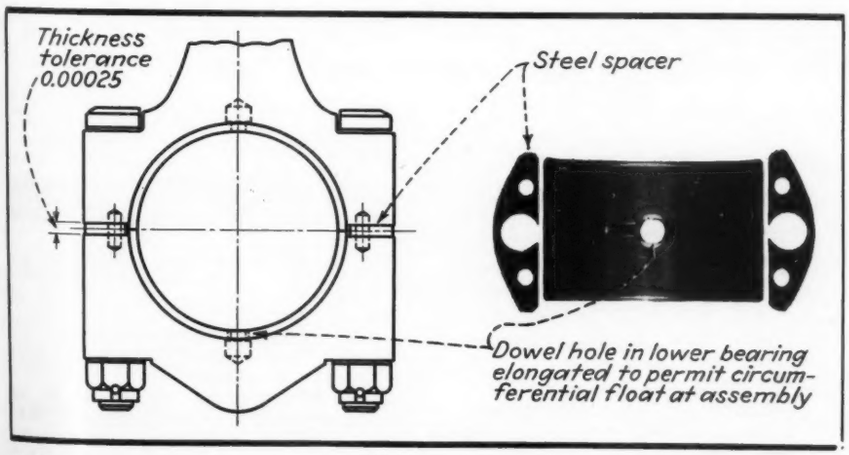

Thickness tolerance 0.00025
Steel spacer
Dowel hole in lower bearing elongated to permit circumferential float at assembly

Fig. 8—Precision insert bearings installed in a connecting rod originally designed for extended shims

FASTENING METHODS FOR

METHODS for fastening flexible materials such as cloth, felt, sheet rubber, leather and insulating material include: sewing, adhesives, vulcanization with rubber, tacking, patented snap fasteners, screw fasteners, riveting, wire stitching, and crimping. Sketches on these pages show typical applications in upholstery, insulation, floor coverings, and for decorative or protective purposes in automobiles, radios, industrial machinery and other products.

Although adhesives and sewing are by far the most popular methods, they are too commonplace to be given much attention here. For information covering general types and properties of adhesives, see "Industrial Adhesives" by John L. Perkins, P.E. April 1939, p. 135-136.

When using other methods, clinch the fastener over as large an area as possible by using washers, grommets, eyelets, or molding strips. Similarly, sharp corners should be avoided, particularly where the material is subject to tension or repeated flexures at the joint. Free edges should be protected to prevent fraying, curling, or catching on pointed objects.

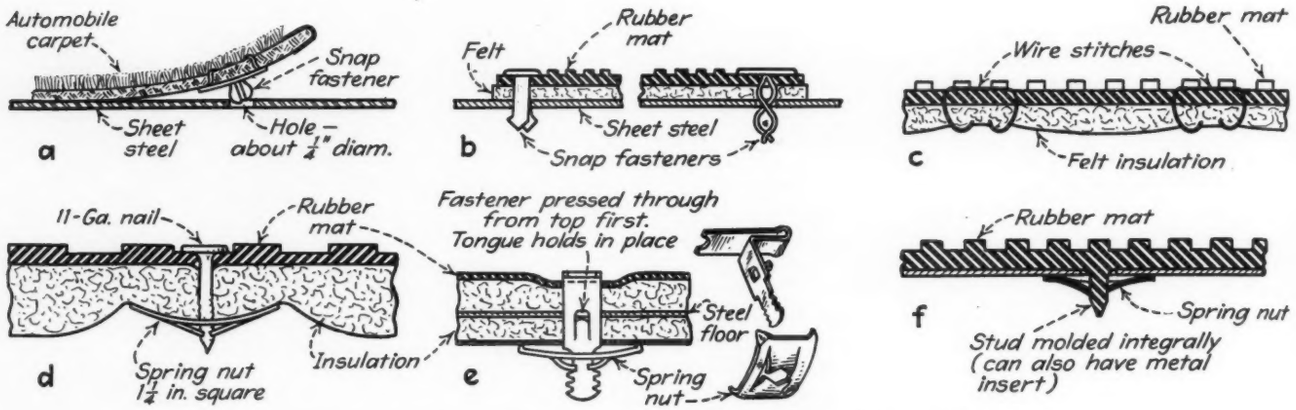


FIG. 1 — FASTENINGS FOR CARPET AND INSULATION

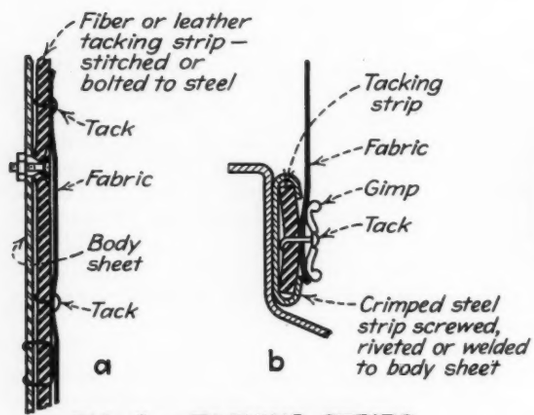


FIG. 2 — TACKING STRIPS

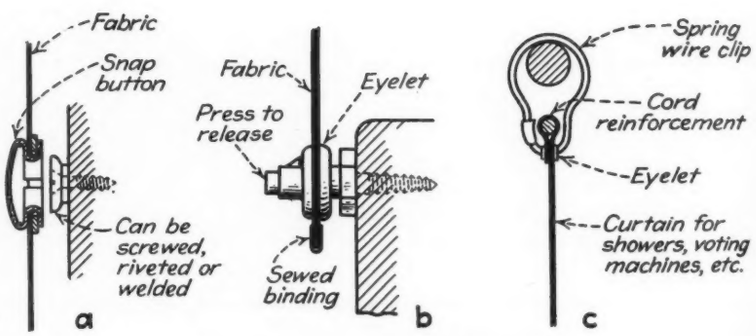


FIG. 3 — SNAP FASTENERS

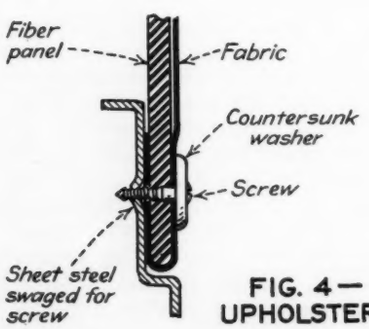


FIG. 4 — UPHOLSTERY

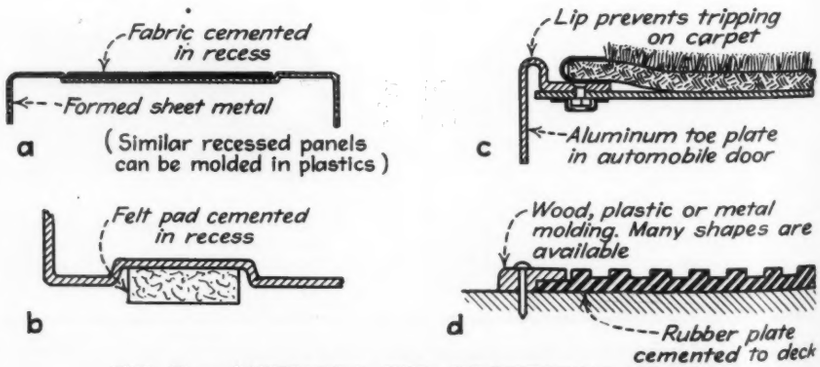

Fabric cemented in recess
Formed sheet metal
a
(Similar recessed panels can be molded in plastics)
Felt pad cemented in recess
b
Lip prevents tripping on carpet
Aluminum toe plate in automobile door
c
Wood, plastic or metal molding. Many shapes are available
Rubber plate cemented to deck
d

FIG. 5 — METHODS FOR PROTECTING EDGES

FLEXIBLE SHEET MATERIALS

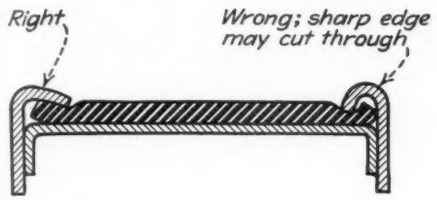


FIG. 6 — CRIMPING OVER RUBBER

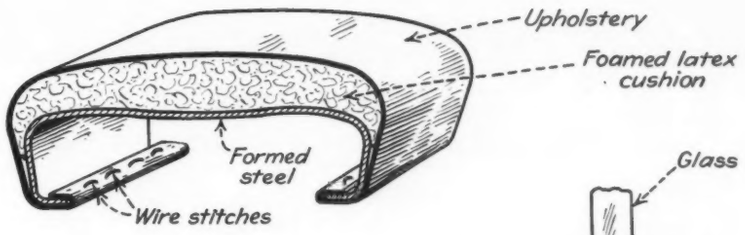


FIG. 7 — CAR SEAT
(Similar method used in card-tables, furniture)

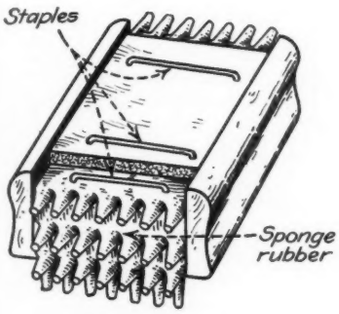


FIG. 8 — SHOE POLISHER

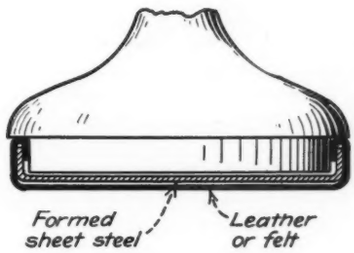


FIG. 9 — TELEPHONE BASE

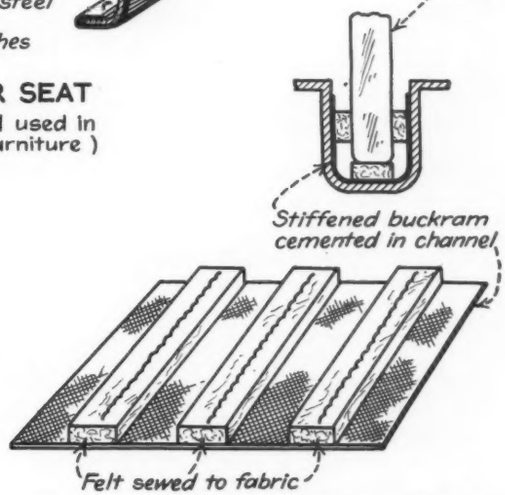


FIG. 10 — FELT WINDOW GUIDE

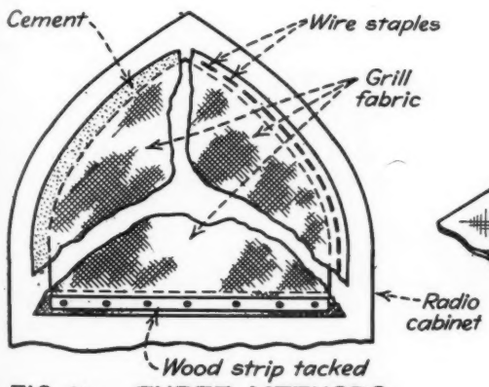


FIG. 11 — THREE METHODS FOR SPEAKER GRILLS

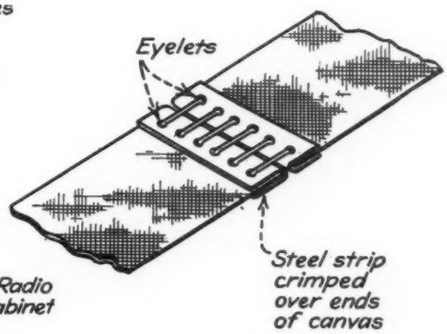


FIG. 12 — CANVAS CONVEYOR BELT

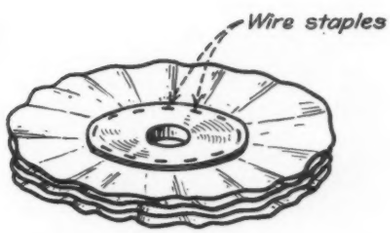


FIG. 13 — BUFFING WHEEL

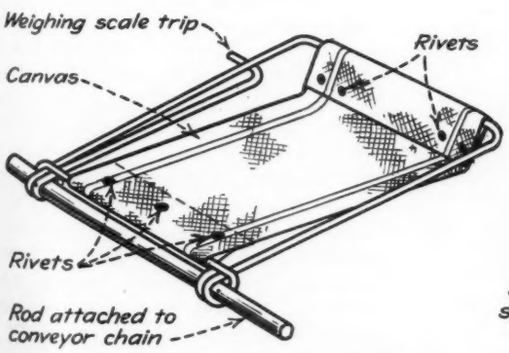


FIG. 14 — CUP FOR APPLE SORTING MACHINE

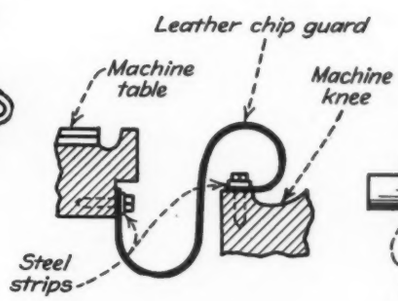


FIG. 15 — APRON FOR PANTOGRAPH

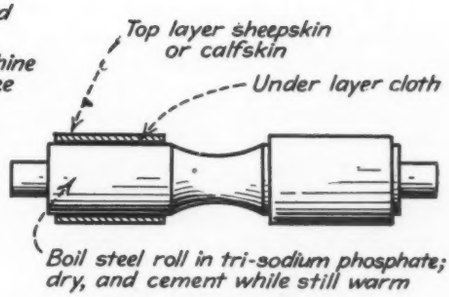

Top layer sheepskin or calfskin
Under layer cloth
Boil steel roll in tri-sodium phosphate; dry, and cement while still warm

FIG. 16 — TOP ROLL FOR SPINNING FRAME

Question and Comment

Flexibility of Generated Gears

W. F. PLUME, *Assistant Chief Engineer*
Philadelphia Gear Works

When gears are form-cut, the rigid rules governing pitch diameter and outside diameter must be followed. The involute form is necessarily modified in these cutters to avoid interference and reduce undercutting. Ranges of teeth which can be cut are specified on each cutter, and in most cases the tooth form is a compromise.

The outstanding advantage of generated gears, besides the higher rate of production and improved accuracy, is that gears may be cut "oversize" or "undersize." The term "long" and "short" addendum is also used. This means cutting standard-depth teeth in a gear blank which is larger or smaller than the standard outside diameter for that number of teeth.

The flexibility therefore modifies the usual conception of gear terminology. Base pitch instead of circular or diametral pitch must be considered as the fixed property and the term "pitch diameter" must be used with discretion. In other words, the base circle diameter is fixed for any fixed number of teeth cut by a hob, regardless of the outside diameter of the gear blank. The pitch diameter is fixed only when a pair of gears are in mesh. The pinion pitch diameter, the point at which pure rolling action takes place, is equal to twice the center distance divided by the sum of the gear ratio plus one. The gear pitch diameter is tangent to the pinion pitch diameter. This holds true for standard gears as well as oversize or undersize gears.

When a straight sided rack form is used in generated gears, undercutting takes place for low numbers of teeth. The main reason, then, for making a pinion oversize was to eliminate this undesirable feature. At the same time other less apparent advantages were gained. The purpose of this discussion is to sum up the improvements and possibilities of non-standard generated gears as compared to "standard" gears.

Fig. 1a shows the serious undercut resulting from hobbing a 12-tooth 14½-deg. full-depth pinion using a standard blank. Outside diameter equals 12 P_d plus two addendums. A form cutter would be somewhat better since the tooth flank is radial inside the base circle. Fig. 1b shows a 12-tooth, 14½-deg. full depth pinion cut in an oversize blank. In this case, outside diameter is 12 P_d plus three addendum.

It can be seen that the involute curve in (b) extends to a greater depth than in (a) and the base of the tooth is wider, thus increasing beam strength considerably. The average radius of curvature of the working surface of the tooth is greater, which increases surface durability.

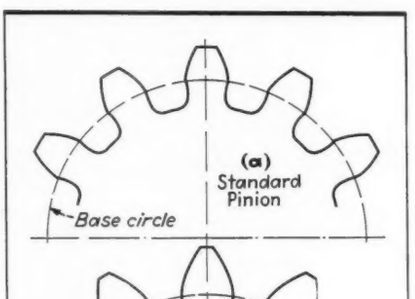


Fig. 1—Comparison between standard and oversize 14½-deg. hobbed pinion

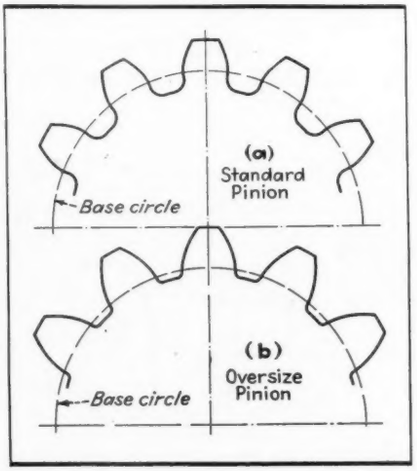


Fig. 2—Comparison between standard and oversize 20-deg. hobbed stub pinion

Fig. 2a shows a generated standard 12-tooth, 20-deg. stub pinion. A form-cut tooth would look very much the same. The stub tooth was originally introduced to reduce undercutting and increase beam strength. Fig. 2b shows an oversize 12-tooth, 20-deg. stub pinion. The oversize pinion has a much better tooth form. The involute curve extends to the bottom of the working depth. Both beam strength and surface durability are improved.

Fig. 1 and Fig. 2 show very clearly the increase of beam strength for the oversize pinion. To get a better picture of what is termed surface durability, look at Fig. 3. For clarity, the tooth outlines are omitted and only the significant diameters and lines of action are shown. Two conditions are shown for a set of gears 12x88 teeth, 20-deg. stub on standard centers. Set *A* is cut with standard outside diameters; tooth contact takes place only along the line marked "*LA–A*," which is the total length of the line of action, as shown in Fig. 3a. Set *B* is the same, except the pinion outside diameter is 12 P_d plus 3 addendums and the gear outside diameter is 88 P_d plus one addendum.

The two lines of action are shown separately in 3b and 3c and contact conditions are illustrated at three points on each. In (b) point (1) is just past the start of tooth action for this standard set. The small circle represents a cylinder with the same radius of curvature as the pinion tooth at that point in contact with an arc representing a large cylinder with the radius of curvature equal to that of the gear tooth. Points (2) and (3) show how rapidly the pinion "cylinder" increases in size as the contact point increases its distance from the pinion base circle.

Fig. 3c shows corresponding contact conditions of the "long and short addendum" set. The load carrying capacity of the larger pinion "rollers" is greater at all points of contact.

Another improvement of set *B* over set *A* is the location of the angle of action. Note that in set *A* the angle of

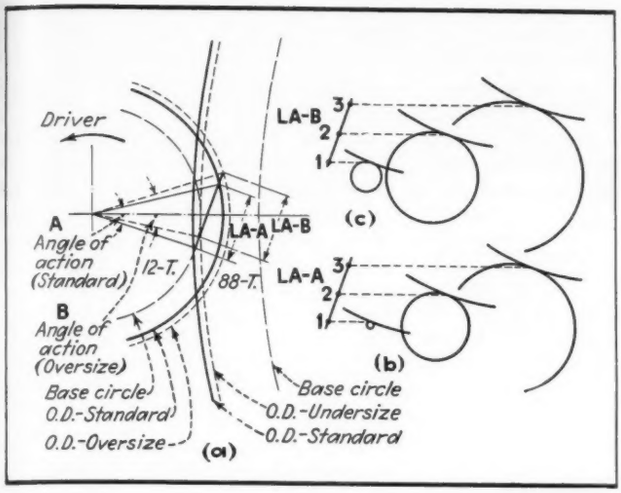


Fig. 3—Significant diameters and lines of action of two sets of 20-deg. stub gears mounted on standard centers. Pinion of one set is generated on oversize blank

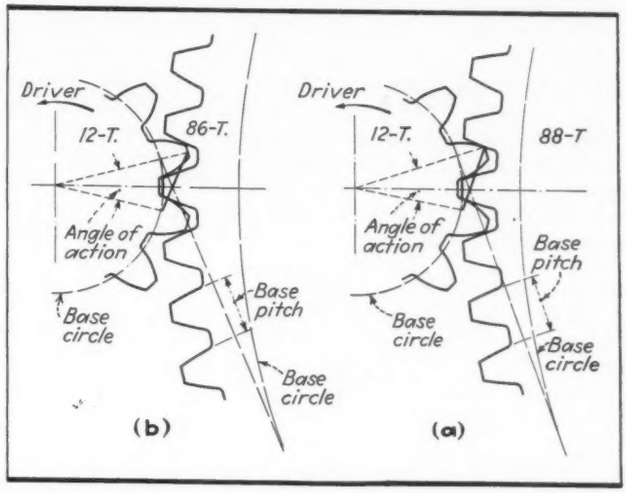


Fig. 4—Diagram of a gear set with standard pinion showing effect of interchanging gears which have different number of teeth but which are generated on same size blank

approach is slightly larger than the angle of recession. During approach rubbing action is toward the base of the teeth. During recession the rubbing is toward the tips of the teeth. The latter action results in less noise and smoother operation.

The discussion thus far has been concerned with the strength, durability and improved action of the oversize pinion as compared to the standard pinion. Now we will investigate the full meaning of the term flexibility as applied to generated gears.

Suppose our machine requires a set of gears on 10-in. centers to give a ratio of 7⅓ to 1. We could use 12 x 88 teeth, 10 P_d, 20 deg. stub as shown in Fig. 4a. The set would be made with long and short addendums as shown to give best performance. Standard gears would fit in for these conditions also, since for 10 P_d, pitch diameter and centers are standard.

Suppose, however, that the same machine was to be designed with an exact ratio of 7-1/6 to 1. A tooth combination of 12 x 86 teeth will give the exact ratio, but on the 10-in. centers the diametral pitch will be 9.8. For form cut gears, special cutters would be required, adding their cost to the total cost of the gears. But, if the gears are generated, they may be cut with a standard 10 P_d hob in oversize blanks. Fig. 4b shows the tooth form when the pinion is exactly the same as in Fig. 4a and the gear is cut in the same diameter blank as in (a). The base pitch is the same in (a) and (b), but the actual circular pitch and pressure angle is greater for (b) than for (a).

These illustrations immediately suggest another possibility. Suppose a machine tool required three speed changes on 10-in. centers with ratios of 7⅓, 7¼ and 7-1/6. These exact ratios are possible in tooth combinations of 12 x 88, 12 x 87, and 12 x 86 teeth. All three combinations may be had using the pinion illustrated in Fig. 4a or (b), changing only the gear. Thus the pinion may be cut integral with the drive shaft and only three change gears need be carried in stock.

The flexibility of generated gears thus permits us first to use standard hobs where special pitches are required to get an exact ratio and second to change the ratio of an existing set of gears slightly by changing only the gear.

When designing non-standard gear sets, certain limitations must be recognized. In the 12 x 88 tooth set it can be seen that the gear is not undercut, even though it is undersize. The pinion, however, would be slightly undercut if made standard and seriously undercut if made undersize. A good rule to follow is to try to get the exact ratio using the standard total number of teeth. If this is not close enough, use less teeth. Too few teeth will result in pointed teeth and incomplete generation of the involute at the tooth flank. Too many teeth for the pitch and centers will result in undercutting. When in doubt it is best to make tooth layouts of pinion and gear generated from a standard hob.

Aircraft Welding Papers Eligible for Two Awards

To the Editor:

I have noted with interest the announcement of the Summerill Tubing Company of the establishment of a series of prizes for papers to advance the art of welding of aircraft steels, including tubing and other steel parts for tubular assemblies.

My purpose in writing you is to point out that papers submitted in the Summerill contest will also be eligible in the $200,000 Industrial Progress Award Program which is being sponsored by the James F. Lincoln Arc Welding Foundation, provided of course, that such papers comply with the Foundation's rules and conditions as regards subject matter and form.

It is conceivable that a paper entered in the Summerill contest would be written in such a way as to be acceptable in the Lincoln Program. If it is not acceptable as written, the author, by a few changes or additions, could make it conform to the Foundation requirements.

—Ed C. Powers
James F. Lincoln Arc Welding Foundation

Author Corrects His Stress Calculation

To the Editor:

Referring to page 242 of your May number and my article on "Composite Members, How to find the Stress in each Material," there is one point where I can improve upon myself.

Upon further study of Case II, which is that of a clad pressure vessel, I have found that this case requires a more specialized form of treatment than that given in the article. This case is quite similar to that of a hooped gun barrel differing only in that the two materials are not shrunk together. It is assumed that any initial stresses due to the cladding process are non-existent. Allowing for this assumption, in the case of a thin cylinder where the ratio of total

thickness to diameter is quite small, the derivation will take the following form:

$T = pr = T_1 + T_2$; $T_1 = p_1 r$; and $T_2 = (p - p_1)r$ very nearly,

where p_1 is an intermediate pressure acting upon both parts conversely along the junction of the two. It is produced by the inner cylinder expanding first under the pressure p and has a magnitude wholly dependent upon the ratio of the respective E's and thicknesses.

$$S_{1a} = \frac{T_1}{t_1} \text{ and } S_{2a} = \frac{T_2}{t_2}$$

$$e_1 = e_2 = S_{1a}E_1 = S_{2a}E_2$$

from which is derived

$$p_1 = \frac{p}{1 + \frac{E_1}{E_2} \times \frac{t_2}{t_1}}$$

and

$$T_1 = \frac{pr}{1 + \frac{E_1}{E_2} \times \frac{t_2}{t_1}}$$

$$S_{1a} = \frac{pr}{t_1 + \frac{E_1}{E_2} \times t_2} = \frac{10{,}200}{\frac{1}{2} + \frac{28}{23} \times \frac{1}{8}}$$

$$= 15{,}600 \text{ lb. per sq.in.}$$

$$\text{and } S_{2a} = \frac{pr}{t_2}\left(1 - \frac{1}{1 + \frac{E_1}{E_2} \times \frac{t_2}{t_1}}\right)$$

$$= \frac{10{,}200}{\frac{1}{8}}\left(1 - \frac{1}{1 + 1.22 \times 0.25}\right)$$

$$= 19{,}000 \text{ lb. per sq.in.}$$

$$T = 19{,}000 \times \frac{1}{8} + 15{,}600 \times \frac{1}{2} = 10{,}175 \text{ lb.}$$

—William I. Powell
City of New York

Test Data Shows Effect Of Water on PVA

To the Editor:

I appreciate very much Mr. Peierls' comments (P.E. May, p. 262) on my paper, "Properties of Synthetic Rubber-Like Materials," and the information he revealed concerning the industrial applications of Resistoflex PVA. As he points out, the term "resistance" is a relative term. If an application can withstand swelling without loss of efficiency, then resistance as measured by loss of tensile strength should be permissible. On the other hand, if the application is adversely affected by swelling, then the percent of swell should be a measure of the resistance.

In connection with the stated limitations of polyvinyl alcohol with respect to water, in my opinion designers of equipment should be acquainted with this situation, though I am in accord with Mr. Peierls that if the amount of water present is of little consequence, and the serviceability is not affected, PVA should be acceptable. However, the dependence of the strength of the resin upon water can be shown by the following test data, taken at 70 deg. F.:

PVA conditioned for 96 hours in atmosphere of:	Shear strength lb. per sq. in.
Dry	8,000
30% humidity	5,700
75% humidity	2,600
Immersed in water	Dissolved

—John Delmonte

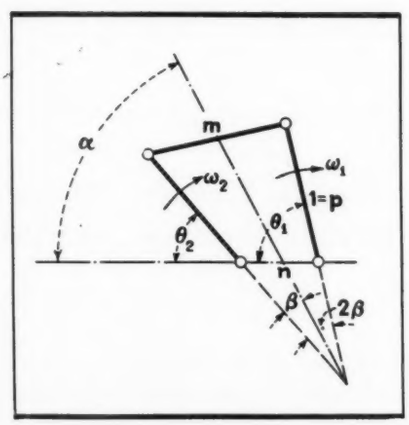
Drag-Link Mechanism Simple in Construction

To the Editor:

The linkage treated in the article "Drag-Link Mechanism," by Albert Shields and Ernest Rosmarin, which appeared in the March number of Product Engineering, page 113, is indeed interesting for its mechanical simplicity. Even if, dynamically, the mechanism is not of the best, it has been used successfully in important applications where usually the maximum (or minimum) speed portion of the cycle is utilized, it being immaterial how the maximum (or minimum) speed is approached. Because of the simplicity of its construction, it is possible that the mechanism will find wider application in the future.

For the particular case $p = 1$, the writer has worked out the following relatively simple method of computation, which might be useful in cases where the speed ratios in the whole cycle are to be plotted.

Referring to the accompanying diagram, let θ_1 be the angular displacement of the driving crank, and θ_2 the angular displacement of the driven crank. Let 2β be the angle included by the cranks, and α the angle that the bisector of 2β makes with the line of crank centers. Then



$$\sin \alpha = \frac{\cos 2\beta - M}{2\, n \sin \beta}$$

where $M = \frac{1}{2}(2 + n^2 - m^2)$, a constant. Now, by means of the above expression different values for α may be calculated by assuming arbitrary values for β; and the corresponding angular displacements of the cranks will be

$$\theta_1 = \alpha + \beta$$
$$\theta_2 = \alpha - \beta$$

The range of variation of β is from β_{min} to β_{max} these limits being determined by the equations

$$\sin \beta_{min} = \tfrac{1}{2}(m - n)$$
$$\sin \beta_{max} = \tfrac{1}{2}(m + n)$$

The second equation corresponds to the position of the linkage in which the cranks are crossed. The speed ratios corresponding to the angular displacements θ_1 and θ_2 as found above may now be computed as follows

$$\text{Speed ratio} = \frac{\omega_2}{\omega_1} = \frac{n \sin \theta_1 + \sin(\theta_1 - \theta_2)}{n \sin \theta_2 + \sin(\theta_1 - \theta_2)}$$

Clearly, the speed ratios may now be plotted in terms of either θ_1 or θ_2 alone.

—J. J. Pesqueira
New York, N. Y.

Aircraft Man Comments On "Castings & Forgings"

To the Editor:

May I call attention to the two articles in Product Engineering by Dr. Ernest Geiger, namely, the one pertaining to Castings and Forgings, the other to Machining. I heartily disagree with the former article (P. E., Feb. 1941, p. 72-73) and commend the latter.

Isn't it true that the examples shown can neither be labeled right nor wrong, without first knowing the specific purpose of each? I am taking exception to the following sketches for the reasons enumerated below.

Fig. 2: It is my opinion that this piece could be placed flat in the mold in such a way that a straight or curved surface would make little or no difference to the pattern-maker or to the foundry-man. Of course, in this case the parting line would be placed on the periphery.

Fig. 3: Stress concentrations built up in the thin-walled section of the counterbore would, I am sure, preclude the possibility of ever obtaining ultimate stress. Secondly, a boss would be a much more desirable type of structure from a cost angle; spot-facing a boss is much less costly than machining a counterbore. Last, but not least, in

these days when speed is paramount, spot-facing would be the only answer.

Fig. 4: Although the basic idea is good, I take exception to the statement that bosses are incorrect. In aircraft and other allied industries, designers must be cognizant of the fact that there is such a thing as weight-saving. Bosses again are the answer.

Fig. 12: The basic idea is well taken, but why the bosses? It is not always wise to vent cavities in castings. I have designed many castings where vents were impractical if left open, due to the fact that moisture collecting in the casting would cause corrosion.

Fig. 14: It is my opinion that neither pattern-makers nor foundry-men would encounter any difficulty in making either part. However, since cost and speed represent dollars and cents in National Defense, the suggested design would probably not be tolerated in any aircraft manufacturing plant because it would require costly engineering time to make two additional drawings, one for the stamping and one for the assembly of the two. Additional costs would include those of making the die for the stamping, the machining of the casting to receive the stamping, to say nothing of the assembly cost. In a contract of one thousand airplanes such assembly would require the stocking of one thousand stampings, which in itself is a cost no aircraft manufacturer would tolerate.

Fig. 16: I believe just the reverse. The part which is labeled as "wrong" is the easier of the two to make because in cutting the male and female dies, concentric machining is used, whereas the other would require the machining of two parallel sides to the tangent point of the small radii. Both examples shown would be forged flat.

Fig. 17: Once again this is an example where no one can say one method is right and the other is wrong. The method employed is entirely contingent upon the length of the finished part. I for one would certainly not swage 18 in. or 19 in. of metal in order to get a 1-in. head, when upsetting is simpler and cheaper.

Fig. 18: I cannot agree that it is at all difficult to forge collars such as shown in the example labeled "wrong." To make a part such as described as "right" would necessitate the very expensive operation of machining a collar to exceptionally close tolerances from bar stock, turning down the small shoulder to equally close tolerances and then sweating one to the other, an operation which is discouraged by both the aircraft branches of the Army and Navy.

—Wm. J. Cushing
Hollywood, Calif.

Can You Work This One?

Solution to May problem—

Did Tilden Win This One?

In a tennis tournament held at Forest Hills, Richards, Cochet, Hunter, Tilden and Johnson were participating. The easiest way to find out who won from the six conditions given is to set up the names of the players across a sheet and the order of finishing down the side. Then, by checking off each possibility as it is eliminated from an examination of the conditions, it is found that the order was: Tilden, Richards, Hunter, Cochet and Johnson.

This month's problem—

Call for Mr. Archimedes!

A large ice cube is floating in a glass just full of water. If the cube is just allowed to melt, will the glass run over, have less in it, or be at exactly the same level as before?

CASE HISTORIES IN PATENT LAW

H. A. TOULMIN, JR.

(Editor's Note: *The purpose of these Case Histories is to give a better understanding of the patent law in the public interest. The author advises that actual cases should be submitted to your own patent attorney.*)

No. 21

Question: What is the effect of war work on filing of patent applications?

Answer: Many persons have the impression that the rule requiring the filing of your patent application within a year from the date of its first public use or sale, or offer to sell, is superseded when the inventor is too busy to file his application because he is engaged in war work. Our Government is accepting no excuses for such a delay.

During the World War, likewise, no excuse of this sort was recognized.

In the case of reissue applications, our Government did recognize that a delay in applying for a reissue, caused by war work, was a satisfactory excuse. It also joined foreign nations by new legislation in extending the period during which applications could be filed in the United States and abroad, where Nationals of the respective countries were legitimately prevented from sending their papers out of their country to the United States Patent Office. But no such excuse exists for citizens of this country.

It is interesting to note that a number of foreign countries in which taxes have to be paid each year in order to keep your patent in force, are passing laws permitting foreigners to delay their payments until these countries become more accessible. In the United States we do not require any taxes to be paid, so this rule does not apply. In some countries like Canada, there are very strict regulations requiring that the nationality and origin, as well as ownership of applications for Letters Patent which are being applied for, be made known to the Canadian Patent Office.

Here again this country has imposed no regulations except through its Export Control. Our country will not permit you to send abroad any information on a long list of technical subjects, ranging all the way from the making of gasoline to the composition of certain alloys, unless you first secure an Export License. Even amendments to existing patent applications, if they contain new subject matter, cannot be sent abroad without securing such a license; and if this rule is violated, the penalty is a very heavy one. This is to prevent the bootlegging to foreign countries of valuable American ideas. Additional legislation is pending in Congress looking to the maintenance of valuable wartime inventions on the Secret List and also directed to preventing any industry from being enjoined under a patent from making wartime materials for the benefit of the Government.

News and Summaries

Raw Materials Shortage May Curb Expansion of Plastics Industry

WHEN OPM REQUESTED that plastics be substituted for strategic metals, it appeared that the plastics industry would receive a tremendous boost. But, according to *Chemical & Metallurgical Engineering* for May, an enormous expansion may not be immediately possible. Reason: shortages of raw materials.

The most serious situation exists in supplies of methanol, from which formaldehyde is derived. Formaldehyde, which accounts for about 20 percent of methanol output, is one of the basic materials for phenolic and urea resins—the plastics most in demand as replacement materials.

Methanol production at present is decreasing rather than increasing in step with the plastics industry. Methanol-producing equipment is being converted to the production of ammonia needed in abnormal amounts for nitric acid. Difficulties in securing equipment for ammonia production may necessitate more change-overs, whereupon the supply of methanol for formaldehyde and other uses will not be sufficient. Scarcity of formaldehyde and methanol has already been reported. A big use for methanol is in anti-freeze for automobiles.

As for phenol, the other basic material needed for phenolic resins, demand is running ahead of supply, and productive activities must be expanded. It has been estimated that 95,000,000 lb. will be required for all purposes this year; production for 1940 was estimated at 75,000,000 lb.

Urea is now a domestic problem, since importations have fallen off. Ammonia is one of the essential materials required for urea and the uncertainty surrounding future ammonia supplies is equally applicable to urea.

Two other raw materials where shortages might occur are cotton linters and nitric acid. The supply of cotton linters is not enough for plastics, rayon and smokeless powder. Powder plants are being urged to substitute wood-pulp alpha cellulose. Nitric acid is needed for explosives.

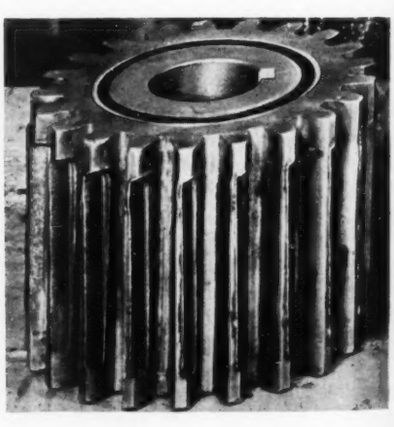
If the shortage of plastics becomes acute, some form of distribution control must be adopted. There are two angles to this situation. First, the strategic importance of plastics must be compared with other industries using the same raw materials. Secondly, the plastics produced must be distributed according to the defense importance of their uses.

Although the plastics industry will undoubtedly expand considerably during the present emergency, the leash will not be off until raw materials diverted from war chemicals become plentiful for use in plastics. But the impetus for this expected expansion is gathering now while new uses for plastics are becoming firmly established.

Inventories Subject To Simple but Rigid Control

INVENTORY control of a new form, designed to prevent the accumulation of excess stocks and supplies, is being applied at once to virtually all metals, E. R. Stettinius, Jr., announced early

300 engineers and draftsmen work in this air-conditioned, acoustically treated drafting room of Westinghouse's new Lester, Pa., office building. Covering entire 300 x 120-ft. second floor, it has glass-brick windows with "islands" of clear glass to prevent feeling of claustrophobia. Continuous-strip fluorescents provide an average of 50 ft.-candles on drawing boards. A lake 2 in. deep on roof will lessen load on air cooling unit during summer; air is electro-statically cleaned. A dumbwaiter saves steps between this room and fireproof tracing vault and blueprint room located downstairs.



Teeth of this bronze punch press drive pinion have worn to half their original thickness, yet the pinion was giving good service in medium heavy automotive stamping operations when removed, according to its user. Drive was from 5-hp. motor at 900 r.p.m. through 18:1 reduction. Pinion, mounted on Torflex rubber bearing, outlasted six fiber pinions, was quieter than steel.

First Tanks Delivered By Three Manufacturers

LATE IN APRIL, three private concerns (American Locomotive, Chrysler, Baldwin) turned over to Army their first-made 28-ton tanks. These were pilot models; line production will get under way as soon as facilities are complete. Chrysler is working on an order for 1,000; American Locomotive will make 685. Figures indicate that medium tanks turned out by August may total 100; several hundred should be delivered this year.

The M-3 combat tank shown (right), 18 ft. long and 8 ft. high, has more than 14,000 parts, some of which require over 20 machining operations. It can cruise at 32 m.p.h.; carries eight .30-cal. machine guns, one .37-mm. cannon and one 75-mm. cannon.

This speedy ponderosity is driven by a fan-cooled engine of 400-plus hp., through clutch, propeller shaft, constant mesh mechanical transmission, differential, and final drive gears to sprockets at front. Design of cooling fan, incidentally, presented a tricky problem. Steering is accomplished by hydraulic braking. Tracks are of forged steel sections to which are vulcanized rubber blocks. Link bearings operate in rubber bushings. Weight of tank is carried by six flexible suspensions, each supported by two rubber-tired wheels.

---

last month from OPM. The plan, he stated, was made necessary by "overbuying for unnecessary inventories." It is part of a coordinated program which for the first time reaches down into all sections of industry.

The control is simple. It provides that shipments of the metals affected may not be made to customers in amounts which would increase the customers' inventories to unnecessary levels. Specifically, after the 10th day of each month, commencing June 10, 1941, no supplier shall make any delivery to any customer unless the customer provide a sworn statement covering inventories received during the preceding calendar month. The supplier must likewise make affidavits to OPM. Forms for these statements have been recommended, and are in suppliers' hands.

Scope: Virtually all metals are affected. The list includes antimony, cadmium, chromium, cobalt, copper, ferrous alloys of all types, iridium, iron and steel products, lead, manganese, mercury, molybdenum, non-ferrous alloys, tin, vanadium. Scrap is included.

Limitations: The plan applies to and is limited to the inventory situation. It does not constitute mandatory, industry-wide allocation such as already has been imposed on aluminum, magnesium, nickel, ferro-tungsten, machine tools, and Neoprene.

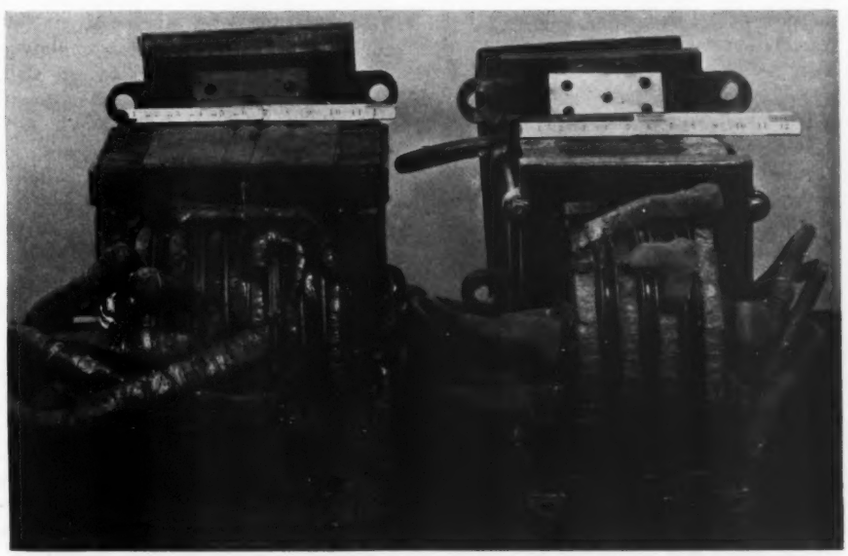
Next step in the coordinated program will be distribution to all industries of questionnaires covering all inventory data. After that will come the withholding of supplies from any users whose inventories are excessive. The Division of Priorities is working on plans for a field staff which will soon make spot checks of inventories.

Emphasis has been placed by Mr. Stettinius on the fact that "no real advantage can be gained by the building up of excess inventories, because such situations will be corrected.

Do You Know That—

HOLLOW-BLADED STEEL propellers for military aircraft will go into production soon. They are lighter and less subject to pitting. (48)

IN TESTING an airplane wing of 100 ft. span, the deflection at the tip of the loaded wing often is 60 in. (49)



These transformers have equal capacity, but the one insulated with Fiberglas (right) is considerably smaller and lighter. Quietly and unobtrusively this evolution is reaching all types of electrical equipment requiring Class B insulation, from tiny 10-watt to giant 40,000-hp. motors.

ENGINEERING IN WASHINGTON

McGRAW-HILL WASHINGTON BUREAU, *PAUL WOOTON, CHIEF*

Washington's best talking subject in the design engineering line as we write, is the confidential report of the National Advisory Committee for Aeronautics on flight characteristics of the Douglas DC-3 transport plane. This report came to light in the current Senatorial investigation of airline accidents. In 1937, NACA was testing low-wing, twin-engine types of planes, civil and military. The DC-3 was new and interesting. NACA asked United Air Lines for a plane and pilot, which were cheerfully given. The ship was tested for five days, and a motion picture was made of the behavior of silk tufts attached to the wing as it approached and went into the stall. Report and picture were submitted at once to airlines and government agencies.

A clean bill of health was given by NACA to the ship for power-off performance, but with power on it was pronounced unsatisfactory. To what extent the DC-3 operators submitted the data to their pilots, or what other action they took, could not be learned prior to their testimony before Congress. But meanwhile, at least three major lines and the Douglas Company, have taken issue with the test methods and findings of NACA, whose technological rating in Washington is top. The Civil Aeronautics Board and the Airline Pilots Association were tacitly in agreement with NACA.

The controversy started when NACA submitted its "confidential" report, which the Committee immediately put in the public record. Discussion will center around possibility that the power-on stalling characteristics of the DC-3 figured in some of the accidents. Ordinarily, when a plane approaches a power stall, there is severe buffeting on the tail members, and the controls go soft. NACA says this warning is seriously lacking in the DC-3.

Tank design requires more precision work than manufacturers expected when they made contracts. Rail car manufacturers, for instance, found that tanks, subjected as they are to the full shock of their own weight, take a much worse beating than cars rolling on steel rails. And, the tough metals used call for difficult tool work. But withal, the rail equipment makers, with their experience in big castings, forgings, and pressings, are best fitted for the job.

Tanks are really designed for light weight—considering the job they have to do. In Panzer warfare, speed and maneuverability are of major importance, and weight curtails both. In fact, the Army is considering seriously whether to go ahead with its "heavy" 50-ton tank for just that reason. Baldwin Locomotive has an educational order, and the project will be thoroughly tried out. Thickest armor on the 25-ton medium tank, is about two inches. This will stop all rifle and machine gun projectiles and shells from 37-mm. guns.

New features being incorporated and developed in the Army's tanks are indirect visual devices, periscopic sights, special padding to protect the crew, cold weather starting, power turrets, electrically controlled guns. Design change will be continuous as it is in aircraft. It is recalled that Americans invented the caterpillar tank, and the British developed the slow, lumbering tank of the last war. It seems probable that the tank will become the equivalent on land of the battleship at sea and the bombing plane in the air.

The reason why aircooled aircraft engines are used in tanks is explained to PRODUCT ENGINEERING by an Ordnance officer: for a high power-weight ratio; because they require small horizontal space; because they use no water, which is sometimes unavailable or strategically unobtainable; because a ruptured liquid-cooling system would cripple the tank. Three types of engines are being used now: The Wright 450 hp., the Continental, and the Guiberson diesel—all radials.

Bomb resisting structure is being studied by the War Department. Findings thus far have been released in a booklet (Government Printing Office, 25 cents) for state, municipal and private engineers. It is not suggested that protection be built now, but only that responsible persons inform themselves. The booklet gives what little is known about the behavior of exploding bombs, suggests factors in bomb-resistance buildings, and discusses air raid shelters. It says the walls of important buildings should be 12 inches thick to resist bomb fragments; five inches of concrete on the roof will stop incendiary bombs of the ordinary types.

Douglas flush riveting process has been released for free use of any airplane manufacturer who wants it. This may result in standardization of riveting processes, of which there are now several in use by various plants, all restricted under patents. The Air Corps is well pleased with the Douglas move. Standardization would save much labor time, would simplify tool procurement, and might simplify maintenance. Heat treating, used in aluminum riveting, is a tricky business.

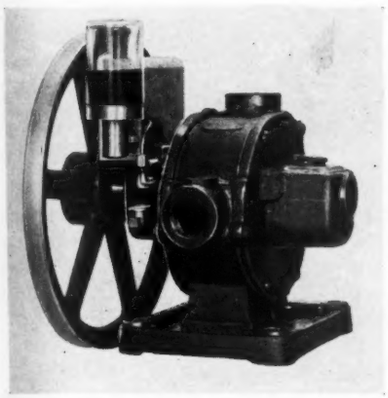
A new steel helmet, similar to that used by the Germans, is being tested by the Army. It probably will be standardized. Principal changes are more protection for the sides of the head and back of the neck. This is accomplished without impairing movement of the head in aiming rifle from a prone position. The helmet weighs 2½ lb.—3 oz. more than the present type. It is made of tougher steel, and is calculated to deflect spent machine gun bullets, shrapnel, and shell splinters, which cause 60 to 80 percent of wounds. About 12 percent of all wounds are received in the head.

About half of the 212 merchant ships authorized recently by the President to be contracted by the Maritime Commission for Britain, will be of the "pointed box" type, and the other half will be of the standard cargo "C" design. The policy of the Government is to build only as many "box" or emergency ships as are absolutely necessary in view of the need for speed. Standard "C" ships are more desirable because: (1) They are faster and are worth about 1½ of the box ships in service; (2) World War experience taught us that emergency ships can't compete on trade routes in peace time, and if they must compete, the subsidy cost is high; (3) "C" designs are not so slow in building as one might think, because materials, design, methods, all were thoroughly standardized through the Commission's educational contracts before the emergency.

New Materials and Parts

Automatic Lubricator

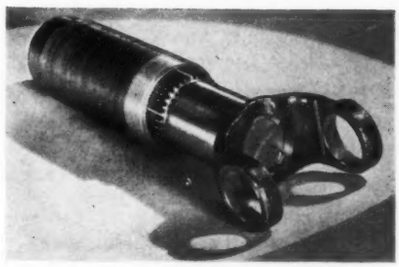
This simple mechanical lubricator which operates only when the machine is operating, is fastened to the machine where a moving part is to be lubricated. Friction contact of the rotating attachment on the lubricator with the revolving surfaces of any shaft furnishes the means of starting and stopping the flow of the oil. Rate of feed of oil can easily be regulated between 1 drop in 10 min. and 10 drops in 1 min. It



does not rely on gravity feed. The automatic lubricator has a visible oil container holding enough oil for a week or more of operation. It is designed to handle light, heavy or medium viscosity oil, is compact in construction, neat in appearance, and inexpensive to buy and operate. Leiman Bros. Inc., 101-FF-7, Christie St., Newark, N. J.

Accurate Splines

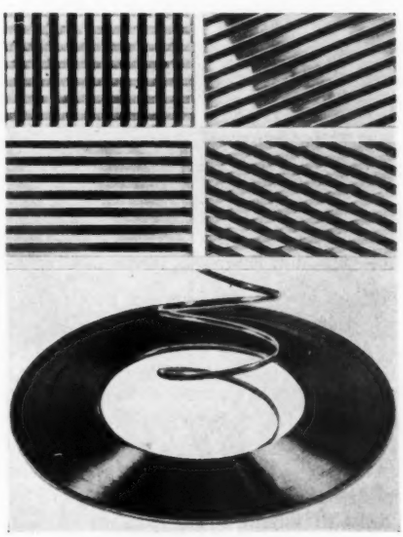
Splines on automotive propeller shafts are now being produced in large scale production by a new process which consists of an adaptation of the "crossed-axis shaving" process extensively used in producing accurate gears. Equivalent accuracy of the female end of the propeller shaft is obtained through the use of high precision broaches. To permit the use of this type of equipment, the splines are actually developed involute teeth. In addition to permitting the use of the crossed-axis gear shaving process, the tooth design provides a stronger as-



sembly, since the increased width at the root of the teeth increases resistance to shear and the increased contact area increases the life of the splines. Furthermore, if wear does occur, noise will be minimized, since any relative movement will be largely in the form of a rolling action, as with involute gearing. Universal Products Corp., Detroit, Mich.

Crimped Designs in Pre-Finished Metals

One of the two new crimp designs in American bonded pre-finished metals is termed a "7/16-in. crimp," and is available in horizontal, diagonal, square, and diamond patterns in a variety of metal thicknesses. Sheet sizes up to 24 x 36 in. are available in most patterns. The second design, termed an "oval crimp," is introduced in two widths—$\frac{1}{8}$ and 3/16 in. in long, continuous coils, and in gages ranging from 0.01 to 0.015 in. Both



crimps are offered in bright or satin finishes of nickel, brass, chromium, or copper electro-bonded to basic metals of steel, zinc, or brass. American Nickeloid Co., Peru, Ill.

Aircraft Motor

This new fractional horsepower explosion-proof aircraft motor meets the requirements of U. S. Air Corps specifications 32159 and 32160 Class "C," Style IV. It has a 1/30-hp. rating at 7,500 r.p.m., and is built for continuous duty, 12 or 24-volt operation. The E2X motor is outstanding for its compact construction, having an overall length of but $4\frac{3}{4}$ in. and weighing but

2 lb. 2 oz. The case is made of magnesium to minimize the weight. It can be either series or shunt wound. Another motor of this series, the A2X is the same as the E2X, except that it is larger, developing 1/15 hp. The Dunmore Co., 14th and Racine Sts. Racine, Wis.

Black Coating for Zinc

An immersion process for the blackening of zinc and zinc alloys, called "Ebonol Z" consists of immersing the zinc or alloy in a 1-lb.-per-gal. solution of Ebonol Z salts at a temperature of from 150 to 212 deg. F. An adherent, jet black finish is formed in from 1 to 5 min. The bath is not critical and requires little control. Unlike painted or enameled coatings, this coating does not change the thickness of the piece more than a few hundred

thousands of an inch; thus close dimensions can be maintained. The solution can also be used for producing a black coating on stainless steel, nickel silver, nickel and the noble metals, by making a couple with these metals and a piece of zinc. The Enthone Co., 442 Elm St., New Haven, Conn.

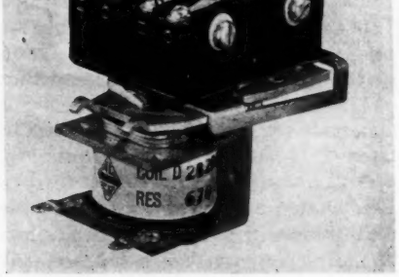
Fractional-Hp. Motor

Built specifically to meet the requirements of machine tools and other industrial applications where frequent start-stop service, plugging, and metal-dust atmospheres are encountered, this fractional-horsepower motor is available in ¼-, 1/3-, ½-, and ¾-hp. sizes for

operation on 3-phase a.c. and d.c. systems. Of totally inclosed construction, it has a sturdy cast iron base, closely machined end-shield and stator rabbets, tough Formex wire windings, ball bearings, a one-piece indestructible cast-aluminum rotor, and firmly anchored windings. General Electric Co., Schenectady, N. Y.

Microswitch Relay

This inexpensive relay is sensitive, with easy acting contacts designed for handling loads up to 10 amp., a.c., with small controlling current. They



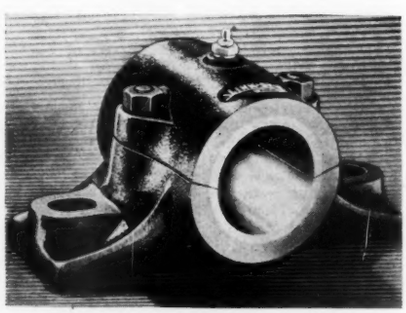
are recommended for locations subject to sudden jarring, vibration, or tilting. The relay, known as the Autelco Jr. Microswitch Relay, is furnished with one or two microswitches, each with make, break, or break-make contacts. Can also be supplied with one break-make microswitch and one break-make spring assembly with code No. 2, No. 4, or laminated silver contacts. Maximum operating voltage is 240 volts d.c., or 50-60 cycle a.c. Automatic Electric Co., 1033 W. Van Buren Ave., Chicago, Ill.

Plastic Coating for Metals

Marlox, a plastic coating for metals, is used either as a priming coat or purely for protection against rust and corrosion. Applied by spraying, painting or dipping, the thin flexible coating is almost completely non-porous, dries quickly, and can be handled 5 to 10 min. after applying. Used to inhibit rust and corrosion, it is applied to a thickness of ½ of 1/10,000 of an inch. As a prime coat, it is applied to a thickness of 5/10,000 of an inch. It can be applied to all types of metals and alloys including steel, cast iron, aluminum, aluminum alloys, magnesium alloys, copper, brass, cadmium plate and galvanized or zinc coated steel. It can also be used to waterproof concrete. Marley Chemical Co., 983 E. Milwaukee, Detroit, Mich.

Flat Box Bearings

Designed for moderate speed and power requirements, this new streamlined babbitted common flat box has maroon-colored, crackle-finish housing. Removable shims between base and cap provide for adjustment. Sloping surfaces between cap and base maintain



concentricity and relieve the cap bolts of direct strain from side thrust. Cap bolts do not protrude above the top of the bearing. Elongated holes for base holding-down provide adjustment for shaft alignment. The babbitt is of a balanced formula, selected in operative tests for maximum service. Link-Belt Co., 307 N. Michigan Ave., Chicago, Ill.

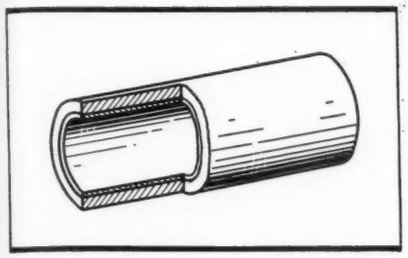
Electromagnetic Relays

These new 4-pole double-throw relays, an addition to the line of Type C relays, feature precise machine assembly of parts and economical quan-

tity production. Self cleaning wiping action of the contacts and long electrical and mechanical life are other features. Operating voltages under normal conditions range from 2 to 230 volts a.c. and 2 to 125 volts d.c. Normal contact capacity is 10 amp. on non-inductive a.c. loads, but special contact materials for specific applications may permit the control of considerably higher current. G-M Laboratories, Inc., Chicago, Ill.

Bimetal Composite Tubing

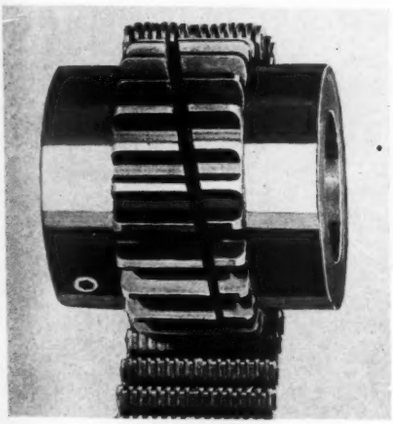
Duplex tubing, having an inner tube of one metal or alloy, and an outer tube of a different metal, has been developed to solve corrosion problems where the corrosion element inside the tube is different from that outside the tube. Bimetallic tubing is now made with the combination of metals chosen to meet these specific conditions. For instance, in oil refinery work, tubing made of steel outside for corrosive oil vapors and copper inside for circulating fresh water has been found excellent; steel outside for same conditions and Admiralty brass inside for circulating salt water. Other combinations successfully used by various industries include: cupro-nickel outside, red brass inside; stainless steel outside, Admiralty brass inside; steel outside, aluminum brass inside. There are many other combinations available for services other



than those described here. A good close contact is obtained between the two materials in the composite tube without interfering with the heat transfer. Where Duplex tubes have to be expanded or rolled into a tube sheet, they are supplied with annealed or tempered ends. Bridgeport Brass Co., Bridgeport, Conn.

Flexible Chain Coupling

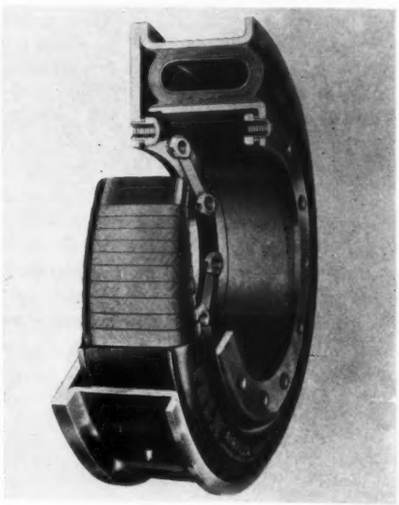
Shearing stresses in the pins of chains for flexible coupling are claimed to be greatly reduced in this new Type D diagonally cut flexible chain coupling. With this coupling, one pinion pulls the other through the chain, employing the full tensional strength of the chain's



width and resulting in the same characteristics as an ordinary chain drive. Tooth wear caused by excessive misalignment is distributed evenly across the entire pinion face. This is claimed to permit greater misalignment and to provide increased life. Ramsay Chain Co., Albany, N. Y.

Airflex Couplings

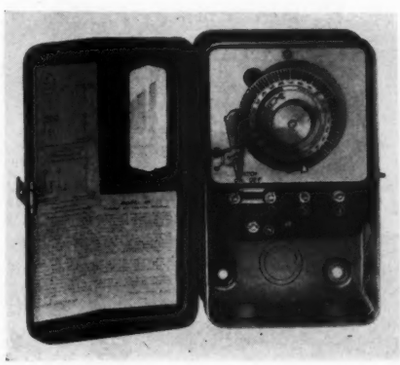
Especially designed for connecting diesel, oil, gasoline, or gas engines to any type of driven machinery, the degree of resiliency of this flexible coupling can be readily changed by varying the air pressure to suit the exact requirements of a particular application. It consists of a special resilient rubber gland, lined with a pre-molded, prevulcanized cylindrical section especially treated to prevent air loss. This cylindrical lining is covered with multiple alternating layers of tough durable fabric and live rubber. The entire gland is permanently bonded to rugged steel inner and outer rims. This frequently eliminates the necessity for adjusting connecting units or making



changes in shaft diameters and lengths or in the flywheel to alter the torsional characteristics of the driving system. Available in six standard types. The Falk Corp., Milwaukee, Wis.

Time Switches

This new 300 Series time switch is a high quality precision instrument which sells at a reasonable price. Only two gears are exposed; all others operate in a sealed oil bath and consequently are always lubricated. It is not affected by dust. Operation is accomplished by slow speed (450 r.p.m.) industrial type self-starting synchronous motor. For actuating the off-tripper, the most difficult operation performed, there is a reserve power of 40 lb. or an equivalent



of 2,000 percent. It operates successfully at temperatures as low as 20 deg. below zero. The case is compact, made of 18-gage steel with a seamless drawn cover and base. Paragon Electric Co., Chicago, Ill.

Print Developing Machine

Known as the Bruning No. 159 Volumatic Developer, this machine is designed to be used with the Model 75 BW Printer. Using this new combination, one operator performs the entire printing and developing operation with ease and speed. Sensitized paper and tracings are fed into the machine at the front, where they are immediately exposed in the printer section. A vacuum separator roll at the discharge point of the printer separates the tracings from the exposed prints, allowing the prints to pass automatically to the developing and drying sections of the developer. The tracing is returned to the operator, while the completely developed print is

delivered flat and dry at the rear of the machine. Developer consists of a separator roll, water roll, and a series of bands which carry the developed prints through the drying section. The developing section is synchronized to operate at the same printing speeds as the printer—speeds ranging from 0 to 25 ft. per min. Charles Bruning Co., Inc., 100 Reade St., New York, N. Y.

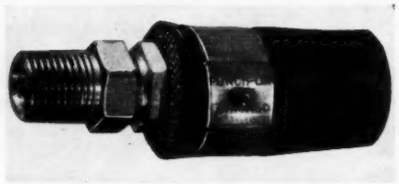
Synthetic Varnish

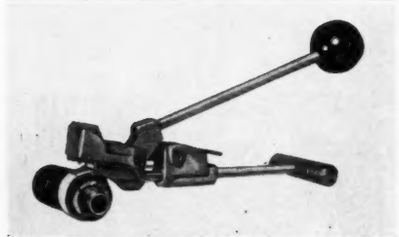
Harvel 612-C is a phenol-formaldehyde synthetic resin-base varnish made from cashew nut shell liquid which polymerizes upon baking to an infusible, insoluble state. The varnish solidifies throughout, leaving no gummy, half-cured interior since it does not depend upon oxidation for its solidification. The new formulation produces a

basically new varnish with improvements in speed of cure and flexibility. It is said to possess unusual retention of insulating strength at high operating temperatures. Irvington Varnish & Insulator Co., 45 Argyle Terrace, Irvington, N. J.

Hose Clamp and Fittings

The Punch-Lok hose clamp, a mechanical device for connecting various kinds of male and female fittings, special nipples, menders or ordinary pipe to a hose, wraps a broad, flat, high-tensile-strength galvanized steel band

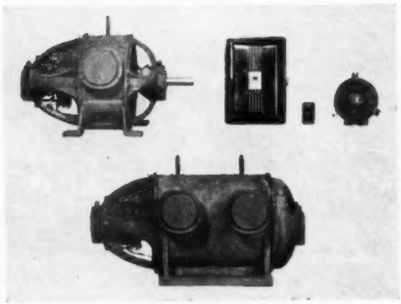




twice around the joint. After tensioning with a pull of 1,000 lb. within the locking tool, the ends are securely locked together without loss of tension within the flat pressed steel clip. Excess band is cut off flush with the clip so that the entire joint is smooth. A single hose clamp is sufficient for all ordinary commercial pressures. Two clamps and a double groove fitting are desirable wherever the fluid is dangerous, such as steam, chemical solutions, or hot liquids. Punch-Lok Co., 430 N. Wolcott Ave., Chicago, Ill.

Adjustable Speed A.C. Drive

This new 10:1 adjustable speed drive, which uses a series circuit without the usual exciter, is designed especially for industrial applications requiring smoothly adjustable speeds over wide ranges with constant torque, in locations where only a.c. supply is available. The new drive has five parts, including control: (1) a single-unit motor-generator set, consisting of a squirrel cage induction motor driving a series d.c. generator which supplies operating voltage for (2) a d.c. series motor coupled to the driven load. In parallel with the generator series field is (3) a



rheostat which controls the driving motor speed. Control apparatus consists of (4) an across-the-line starter for the squirrel-cage motor, and (5) a push-button station. The new drive is more flexible than the wound-rotor motor and is more efficient than the conventional variable-voltage system because it has no exciter rotational losses. Available in ratings from 1 to 15 hp. with a standard speed range of from 175 to 1,750 r.p.m., for two or three-phase operation on 220, 440, 550 volts, 60 cycle systems. Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa.

Foot Switch

Adaptable to all types of machines, this new Smith-Swich responds to the slightest down pressure of the foot and breaks the circuit instantly when the foot is raised. With rubber or neoprene case, Bakelite interior, no exposed metal, no soldering, and ¼-in. dia. tungsten points and phosphor-bronze spring, this switch is guaranteed for 40,000 operations per year. Neoprene unit is moisture proof, dust proof and resistance to oils and chemicals. Units are easily disassembled by snapping off the casing. Nathan R. Smith Mfg. Co., 105 Pasadena Ave., South Pasadena, Calif.

Immersion Thermostats

Designated Series 80, these new immersion thermostats are available in the three following temperature ranges: Model 80L from −50 to 350 deg. F.; Model 80M from 150 to 650 deg. F.; and Model 80H from 350 to 1,100 deg. F. All units are of the metal expansion type with fixed stem and feature the use of dual coarse and micrometric adjustments to facilitate accurate settings of critical temperatures. Excellent temperature regulation is assured with differentials averaging 1 percent and less. Electrical capacity is 2,150 watts, non-inductive load at voltages up to 650 volts a.c. Inductive loads up to ½ hp.

can be handled at voltages up to 250 volts a.c. The switch mechanism is of the positive snap-action type with large fine silver contacts. Connections can be supplied normally open, normally closed, or three-wire. All Tempostats are finished with sprayed aluminum on Bonderized steel. Internal parts are rust-proofed. D. & M. Mfg. Co., 51-53 Lincoln Ave., Midland Park, N. J.

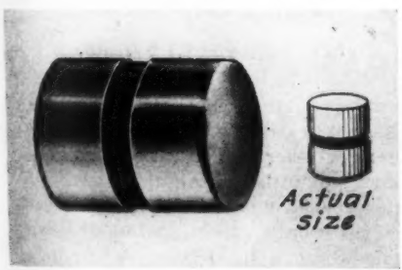
Totally Inclosed Motor

New improved totally inclosed fan-cooled motors are recommended for installation where the air is foggy with metal-cutting solutions, or where there are abnormal quantities of metallic, abrasive and other dusts in the atmosphere that would ordinarily injure or clog the windings of a general purpose open motor. A generous quantity

of cooling air is forced through the large air passages by the non-sparking fan, causing a cooler motor with cleaner air passages. Air intake passages are not easily clogged and are so designed for safety that a $\frac{5}{16}$ in. rod will not pass through them. A steel deflector assists in air-cooling the bearings. Century Electric Co., St. Louis, Mo.

Small Mercury Switch

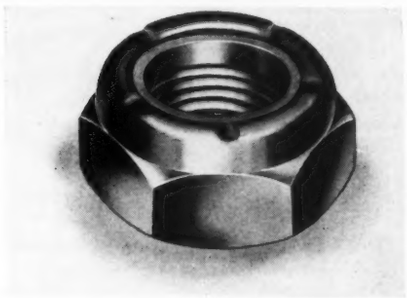
Extremely small mercury switch measuring only $\frac{7}{16}$ in. long by $\frac{3}{8}$ in. dia. is designed for use in low voltage circuits up to 25 volts a.c. or d.c. and currents up to 10 amp. at 6 volts, and 3 amp. at 25 volts. There is no friction or wear in operation. Mercury is contained in the durable metal and Bakelite body. A newly designed baffle device assures positive make or break operation, with no flickering action when equipment is jolted. Suggested uses are thermostat circuits, automobile



glove compartments, trunks and hood lights, radio door lights, indicator circuits, and similar applications. Littelfuse, Inc., 4748 N. Ravenswood Ave., Chicago, Ill.

Thin Hex Nuts

This improved line of self-locking thin hex nuts have approximately 40 percent of the strength of standard-height hex nuts and have been developed to meet the demand for a self-locking fastening which offers savings in space requirements, weight, and cost. As in the standard-height Elastic Stop Nuts, the self-locking action is accomplished by means of a vulcanized fiber collar which is built into the head of the nut. This tough bone-like material resists the entry of the bolt, thus forcing the nut outward and taking up all



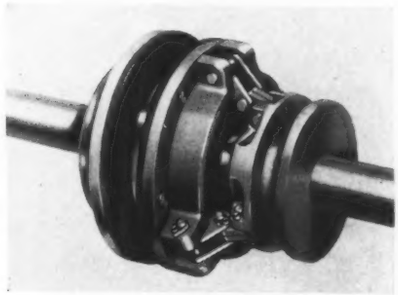
thread play. The nuts are available in steel, brass, and aluminum, in a complete range of standard sizes, both coarse and fine thread. Elastic Stop Nut Corp., 2332 Vauxhall Rd., Union, N. J.

Photoelectric Relay

For controlling electrical circuits in accordance with the rise and fall of natural illumination, the user of this Sun-Switch photoelectric relay chooses the two lighting levels at which he wishes the load switched on and off, and adjusts the calibrated dials to the corresponding foot-candle readings. Operation is entirely automatic, no resetting being necessary. Rugged housing is of weatherproof drawn metal. Operation is from 110 volts, 50 or 60 cycles a.c. United Cinephone Corp., Torrington, Conn.

Clutch

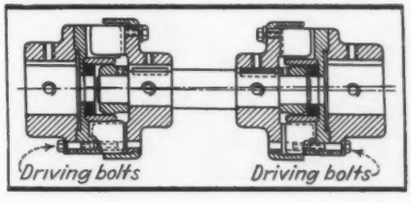
For use with gasoline or other motors up to 6 hp., this compact, powerful, durable clutch has uniformly distributed driving pressure applied directly opposite facing, where it is most effective. The clutch transmits 1 hp. at 100 r.p.m. Clutch plates are steel, hardened and ground. High grade, specially selected facing material is securely fastened in place. Operating links, pins, and rollers are all of hardened steel. Toggle action goes overcenter, locking the clutch in the driving position and providing easy engagement. No oiling is necessary for the



accurate metal bushing which carries the driven sprocket, pulley, sheave, flange, or other parts of machine in which clutch is installed. Rockford Drilling Machine Div., Borg-Warner Corp., 1315 Eighteenth Ave., Rockford, Ill.

Flexible Coupling

This new coupling, designed L-R Type HKQ, is applicable to either horizontal or vertical drives, and is recommended for drives where the space between the driver and driven units cannot be successfully handled by standard couplings, and where there is excessive misalignment. As shown, the floating shaft is supported in fixed bearings, entirely independent of the power-transmitting elements which are thus entirely unaffected by forces generated by the floating shaft itself. Chatter caused by end-wise impact of the floating shaft against the fixed shafts is prevented by suitable cushions. Three types of cushions are used: a long-wearing brake lining material for heavy loads, leather for sustained loads



with infrequent or no cyclic variation. and vulcanized duck and rubber material for fluctuating loads, the absorption of vibration, and maximum misalignment. Lovejoy Flexible Coupling Co., 5009 West Lake St., Chicago, Ill.

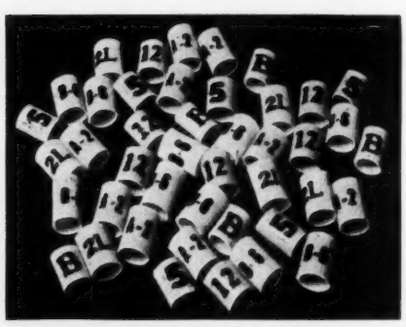
Thermal Oiler

Visible, unbreakable bottle oiler automatically lubricates solid, wick, or waste-packed bearings. On slight temperature rise, the oiler discharges a few drops of oil. As soon as the bearing receives this oil, it cools and the feeding

stops automatically. Adjustable feed is regulated by turning the thermo-dome which opens or closes a port hole. All metal parts are of solid brass, bright cadmium plated. Made in 1, 2 and 4 oz. capacities. Trico Fuse Mfg. Co., 2948 N. 5th St., Milwaukee, Wis.

Ignition Markers

These improved ignition markers, suitable for aircraft engines and any other industrial applications with similar requirements, consist of lengths of inside-and-out varnish-impregnated



tubing, on which suitable identification is screen-printed with a specially formulated ink. This latest form of marker, which is slipped over the wire, is an improvement over the former type in which the identification numerals and letters were printed on cellophane strips which were fastened to the insulating tubing. Available in two diameters, nominal inside diameter of 0.263 or 0.294 in., with either of two heights of symbols, and in two standard lengths. Irvington Varnish & Insulation Co., 24 Argyle Terrace, Irvington, N. J.

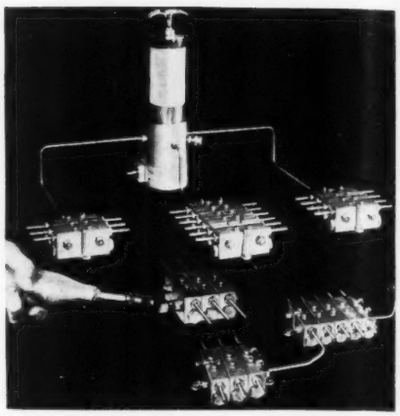
Speed Control Valve

This new two-direction speed control valve controls the speed of piston travel. Installed between operating valve and one end of a cylinder, one valve provides adjustable control of inflow as well as exhaust of the air independently to and from one side of the piston. One valve will, therefore, control the piston speed in two directions. For extra sensitive adjustment and control two control valves, one for each end of the cylinder, are recommended. The valve is so constructed that the two adjusted orifices which control the airflow are set before flow takes place. This insures control from the very start

of movement. Valve body is cadmium plated and all other valve parts are made of corrosion resistant materials. Recommended for 250 lb. per sq. in. maximum air pressure; available in $\frac{1}{8}$, $\frac{1}{4}$, $\frac{3}{8}$, $\frac{1}{2}$, $\frac{3}{4}$, and 1 in. pipe sizes. Hanna Engineering Works, 1765 Elston Ave., Chicago, Ill.

Centralized Lubrication

Development of a non-reversing single inlet, multi-outlet distributor feeder and a new variable speed multi-outlet pump make possible this new non-reversing single pipe centralized lubrication system. The improved distributor consists of a bank of three or more sections each of which discharges a measured quantity of lubricant alternately through one or two discharge outlets which are direct-connected to bearings. By selecting the proper number and capacity of sections, and sup-



plying the proper amount of lubricant to the inlet, a single distributor discharging progressively through one outlet after another will deliver just the desired amount of lubricant to all the connected bearings. The new Trabon Series MP multi-outlet pumps, designed especially for use with these non-reversing systems, are available in three sizes having different reservoir capacities. Trabon Engineering Corp., 1814 E. 40th St., Cleveland, Ohio.

Lubricating Oil for High Temperatures

Unlike conventional petroleum lubricants which often begin to break down or crack at 400 deg. F., and thereby leave a residue of carbon, varnish or other matter which coats bearings and causes wear and power loss, this new oil known as Caloria is recommended only for temperatures above this where it lubricates. While doing so Caloria oil evaporates completely, leaving the lubricated surfaces absolutely clean. Extensive field tests have proven that this "vanishing oil" is the solution to hot spot lubrication such as is found in kiln cars, glass-making machinery, ceramics and glass molds, working parts of die-casting machines, and various hot parts of machines in the metal industries. Available in several viscosities. For use in those cases where Caloria cannot be reapplied before complete evaporation takes place, colloidal graphite has been added—a mixture known as Van Caloria. Colonial Esso Marketers, 26 Broadway, New York, N. Y.

Manufacturers' Publications

Materials

AMPCO METAL—Ampco Metal, Inc., Milwaukee, Wis. Cardboard slide selector, 4 x 8½ in. Physical properties and chemical composition of all grades of Ampco Metal can be quickly ascertained by manipulating this cardboard slide.

COPPER—Revere Copper and Brass, Inc., 230 Park Ave., New York, N. Y. "Revere Copper and Copper Alloys for the Mechanical Industries," 44 pages, 8½ x 11 in. General treatise of chemical and physical properties of the various copper alloys.

FABRICATED MASONITE—Service Industries, Inc., 2025 So. Calumet Ave., Chicago, Ill. "Fabricating to Specifications with Industrial Masonite," 16 pages, 8½ x 10½ in. Illustrates and describes the various types of shapes and forms that can be made from Masonite by bending, die cutting, punching, shaping, routing, laminating and finishing.

RESIN GLUE—Plaskon Co., Inc., 2112-24 Sylvan Ave., Toledo, Ohio. "Plaskon Resin Glue," 12 pages, 8½ x 11 in. Well-illustrated booklet sets forth features, advantages, and typical applications of resin glue and plywood.

SYNTHETIC RUBBER—B. F. Goodrich Co., Akron, Ohio. "Ameripol" 8½ x 11 in. Detailed description of the properties of Ameripol as they relate to mechanical rubber goods products is given in this bulletin.

TUBES AND PIPES—Babcock & Wilcox Tube Co., Beaver Falls, Pa. Technical Data Cards No. 103-A and 107-A. Seamless tubing wall thickness data in millimeters and inches, and a revised list of standard specifications for seamless tubes and pipe arranged in handy fingertip form.

Mechanical Parts

BASE PLATES—Link Belt Co., 307 N. Michigan Ave., Chicago, Ill. Booklet 1882, 8 pages, 8½ x 11 in. Covers the company's line of welded steel base plates for adjusting pillow blocks and common flat boxes for shaft alignment.

LINDSAY STRUCTURE—Dry-Zero Corp., Lindsay Structure Div., Merchandise Mart, Chicago, Ill. Technical Booklet, 16 pages, 3¾ x 6 in. Explains the basic Lindsay structure and principle, describes its development and gives results of laboratory and field tests.

METAL PRODUCTS—L. S. Grammes & Sons, Inc., 362 Union St., Allentown, Pa. Catalog 66, 64 pages, 8½ x 11 in. Illustrates and describes literally thousands of small stamped, drawn, spun, wire-formed, enameled and embossed metal products.

MINIATURE BEARINGS—Miniature Precision Bearings, Split Ballbearing Corp., Lebanon, N. H. Bulletin 3P, 4 pages, 6x9 in. Describes the new self-aligning pivot ball bearing, a new addition to the line of miniature precision bearings.

VARIABLE-SPEED CONTROL—Reeves Pulley Co., Inc., Columbus, Ind. Bulletin, 6 pages, 8½ x 11 in. Describes hydraulic automatic controls for automatic synchronization of machines, automatic speed control, and automatic tension control.

V-BELT DRIVE—Fort Worth Steel & Machinery Co., Fort Worth, Texas. Data book, 64 pages, 8½ x 11 in. Comprehensive data book covers all types of V-belt drive applications, and contains full and detailed information concerning the widely used "Boltrim" or demountable hub sheave.

Electrical Parts

ELECTRICAL CONTACTS—Gibson Electric Co., 8350 Frankstown Ave., Pittsburgh, Pa. Catalog C-10, 14 pages, 8½ x 11 in. Sets forth physical and electrical features of Gibsiloy silver alloy electrical contacts. Many interesting typical applications are illustrated.

HEATING CABLE—General Electric Co., Schenectady, N. Y. Bulletin GEA-3539, 4 pages, 8 x 10½ in. Illustrates, describes and gives engineering data for electric heating cable.

INDUSTRIAL CONTROL MOTORS — The Brown Instrument Co., Wayne & Roberts Aves., Philadelphia, Pa. Catalog 77-1, 22 pages, 8 x 10½ in. Describes a line of motor power units and motorized valves designed to operate with Brown Control Instruments. Photograph illustrations, dimension tables and schematic diagrams are included.

LIMIT SWITCHES—Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa. Descriptive data 15-075, 8½x11 in. Application and construction of 2, 4 and 6-pole planer limit switches for the machine tool industry are discussed. Features of the switch are clearly shown.

MOTOR CONTROL—General Electric Co., Schenectady, N. Y. Bulletin GES-2456, 12 pages, 8 x 10½ in. Illustrates many applications of General Electric combination starters for industrial motor control.

MOTOR CONTROL—Square D Co., 6060 Rivard St., Detroit, Mich. Digest 126, 76 pages 8½ x 11 in. Condensed catalog illustrates and describes safety switches, service equipment, multi-breakers, circuit breakers, panel-boards, motor controls and pressure switches.

PILOT LIGHT ASSEMBLIES—Drake Mfg. Co., 1713 W. Hubbard St., Chicago, Ill. Form 4, 4 pages, 8½ x 11 in. Complete listing and description of dial and jewel pilot light assemblies with information on voltage, type, and mounting.

RELAYS AND SWITCHES—R-B-M Manufacturing Co., Logansport, Ind. Catalog 1941, 34 pages, 9 x 11½ in. Leatherette-bound catalog gives complete data covering relays, switches, pilot lights, harness, and other electrical devices, including packaged units.

SELSYNS—General Electric Co., Schenectady, N. Y. Bulletin GEA-2176, 8 pages, 8 x 10½ in. General treatise on Selsyn self-synchronous apparatus for remote signaling, control, and indication. Applications, operation, features and general types available are illustrated and described.

Finishes

FINISHING WITH ABRASIVES—The National Metal Abrasive Co., 3560 Norton Rd., Cleveland, Ohio. Bulletin A-221, 20 pages, 5½ x 8¼ in. Well-illustrated general treatise can best be described by its title, the "ABC and XYZ of Cleaning, Hardening, Surfacing, Relieving Metal Fatigue and Strain by Controlled Abrasives".

INFRA-RED FINISHES—Ault & Wiborg Corp., 75 Varick St., New York, N. Y. "The Facts About Infra-Red," 10 pages, 5¼ x 7¾ in. Reprint of a magazine article by M. M. Wilson describing the application of finishes to infra-red operations.

WOOD GRAIN REPRODUCTION—Ault & Wiborg Corp., 75 Varick St., New York, N. Y. Bulletin, 4 pages, 8¾ x 11¼ in. Describes an improved method of reproducing wood grain on metal and wood surfaces.

Miscellaneous

DRAFTING MACHINE—The Drafto Co., 233 Walnut St., Cochranton, Pa. Bulletin 141, 4 pages, 8½ x 11 in. contains information as to construction, operation details and prices of the "Master-Drafto" Model No. 60 drafting machine.

ENGINEERING JOURNAL — Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa. This new quarterly journal of engineering information, on the generation, transmission and distribution of electric power will be distributed to a selected list of engineers and executives, and will also be available by subscription.

SOUND LEVELS—De Bothezat Ventilating Equipment Div., American Machine and Metals, Inc., East Moline, Ill. Form FN 4-41, 12 pages, 8½ x 11 in. This booklet is written in non-technical language on the subject of sound levels and other related factors governing selection of axial flow fan units.

Books and Bulletins

Electro-Magnetic Devices

HERBERT C. ROTERS—*561 pages, indexed, 6½x9¼ in. Blue clothboard covers. Published by John Wiley and Sons, Inc., 440 Fourth Av., New York, N. Y. Price $6.*

Intended primarily as a textbook for graduate students, this book will also serve as a useful reference for engineers working on development of new magnetic devices. The book, which assumes the reader is familiar with basis subject matter, is divided into two sections: the first covering theory and methods applicable to all types of magnetic circuits and non-rotary electromagnetic devices, and the second devoted to design of such devices as tractive magnets, time-delays, high-speed magnets, a.c. magnets and relays. The book is based on the author's graduate course at Stevens Institute.

Plastics in Industry

PLASTES—*241 pages, indexed, 39 illustrations, eight diagrams, 6x8¾ in. Blue clothboard covers. Published by Chemical Publishing Co., 236 King Street, Brooklyn, N. Y. Price $5.*

Discussing manufacture, molding technique and applications of the more common plastics, this well-written British book covers the field rather well. A definite effort has been made by the anonymous authors to point out advantages and limitations of each type.

The seventeen-chapter book is well illustrated with recent applications and contains considerable tabular data on performance of materials. Separate chapters are devoted to the application of plastic in the electrical, motor car, textile and aircraft industries. A chapter on synthetic rubber is also included.

Patent Fundamentals

LEON H. AMDUR—*305 pages, 80 illustrations, 6x8¾ in. Blue clothboard covers. Published by Chemical Publishing Co., 234 King St., Brooklyn, N. Y. Price $4.*

Many brief case histories make this book on patent fundamentals extremely easy to read, and dramatically demonstrate nearly all the points Mr. Amdur emphasizes as being of value to design engineers and inventors. The book develops rapidly from a general discussion of the nature of and types of patents to a step-by-step outline of the procedure for preparing and prosecuting patents.

The author devotes considerable space to a discussion of what constitutes invention, and shows how court rulings in interesting cases have determined the accepted interpretations.

Design of Machine Elements

VIRGIL M. FAIRES—*Revised edition, 490 pages, 389 illustrations, 6¼x9¼ in. Brown clothboard covers. Published by The Macmillan Co., 60 Fifth Ave., New York, N. Y. Price $4.*

This revised edition of Mr. Faires' excellent textbook brings up-to-date information in fields where progress has been rapid, and adds a new chapter on variable stresses and stress concentrations. Other improvements are the consistent use of dynamic loading as the basis for the design of all types of gears, and the adoption of the hydrodynamic theory as the approach to the design of thick-film journal bearings. Additional data on the properties of materials and more information concerning the computation of stresses in welded connections are included.

Supplementing the book, a 147-page pamphlet of "Problems on the Design of Machine Elements" presents problems for practice or quizzes on all the important points brought out in the textbook. Price is $1.40.

• • •

Fabrication of Stainless Steels

28 pages, 8½x11 in. Published by Allegheny Ludlum Steel Corporation, Pittsburgh, Pa.

Workability and fabrication of the stainless steels can best be discussed by dividing the steels into two general classifications: (1) austenitic steels which are composed chiefly of chromium, nickel and iron, are non-magnetic, cannot be hardened by heat-treatment, are relatively stiff in annealed condition, but are extremely ductile; and (2) straight-chromium-iron alloys which are intensely air hardening with from 4 to 14 percent chromium, which air harden to a much less extent with from 14 to 18 percent chromium, and which do not air harden to any appreciable extent from 18 to 30 percent chromium. A general discussion of fabrication, finish and heat-treatment of all of these alloys is given in this booklet.

Tolerances for Cylindrical Fits

JOHN GAILLARD—*24 pages, 17 illustrations, 7¾x10¾ in. Published by American Standards Association, 29 W. 39th St., New York, N. Y. Price 25 cents.*

The designer's problem is to specify the required degree of tightness for cylindrical fits in such a way that the shop can establish them. Mr. Gaillard in his booklet describes the basic principles underlying a standard system of fits and tolerances, beginning with the assumption that shafts and holes can be made exactly to specified sizes and progressing from this idealized case to the practical problem of tolerances in a standard system of fits. He lucidly describes and criticizes existing national and international standards. Nothing is said in this booklet about the selection of correct type of fit for a given application—that is left up to the designer.

Machining Zinc Alloy Die Castings

32 pages, 8½x11 in. Published by the New Jersey Zinc Company, 160 Front St., New York, N. Y.

This greatly-enlarged edition of "Practice in Machining Zinc Alloy Die Castings" now includes general data and illustrated examples of every method for machining and forming die castings. Recommended reading for product engineers are the sections on swaging, riveting, and bending and forming, which present ideas for simpler die-casting design.

Welded Steel Tubing Handbook

86 pages, 8¾x5⅝ in., leatherette ring binder. Published by Formed Steel Tube Institute, 1621 Euclid Ave., Cleveland, Ohio. Price $1.

This handbook is an excellent achievement of its purpose—to give the engineer a knowledge of and reference for the physical, chemical and metallurgical properties of welded carbon and alloy steel tubing together with commercial tolerance limitations and other extensive engineering data. In addition to the tabulated data covering commercial sizes, tolerances, and properties, many pages of sketches illustrate typical shapes, fabrication methods, and correct ways to specify dimensions.

Deep Drawing, Shearing and Perforating Of Monel, Nickel and Inconel

Bulletin T-19, 12 pages, 8½x11 in. Published by Development and Research Div., International Nickel Company, Inc., 67 Wall St., New York, N. Y.

Well illustrated bulletin contains directions for shop practices that have proven successful for blanking, cupping, drawing and other operations encountered in press forming nickel-copper alloys, commercially pure nickel, and nickel-chromium alloys.

CHAIN DRIVE LENGTHS

JOHN SCHAPPI
Morse Chain Company

The simple self-explanatory chart, Fig. 1, can quickly be applied to calculate the approximate chain length in pitches if the number of teeth in the sprockets, the center distance, and the chain pitch are known. The formula used with this chart is:

$$L_p = \frac{T+t}{2} + 2C_p + A \qquad (1)$$

where

L_p = chain length in pitches
T = No. of teeth in large sprocket
t = No. of teeth in small sprocket
C_p = center distance in pitches
A = determined from chart

The number of teeth in the sprockets necessary to give the desired speed ratio can quickly be selected by referring to the chart, Fig. 2.

The following new formula can be applied where extreme accuracy is required:

$$L_p = \frac{T+t}{2} + 2C_p + K\,(T-t) \qquad (2)$$

where K is a constant calculated for various values of $(T-t)/C_p$. On page 335 are tabulated values of K accurate to six decimal places.

This new formula is considerably easier to apply than the previously accepted formula which differs only in its last

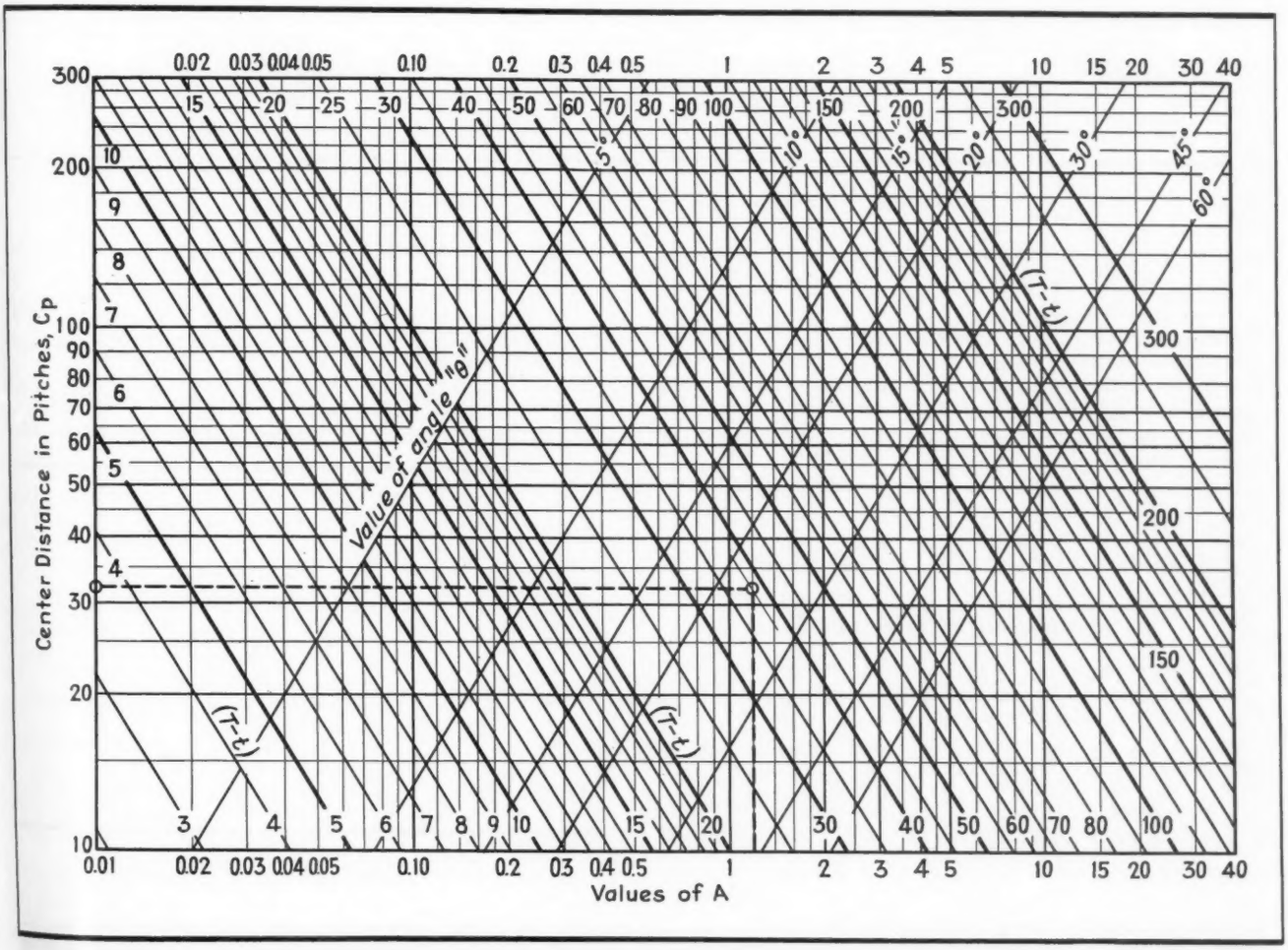


Fig. 1—Chart for quick determination of term A in Equation (1). Enter chart at left with C_p, move horizontally to intersection with diagonal corresponding to (T-t), then move down vertically and read A in pitches at bottom

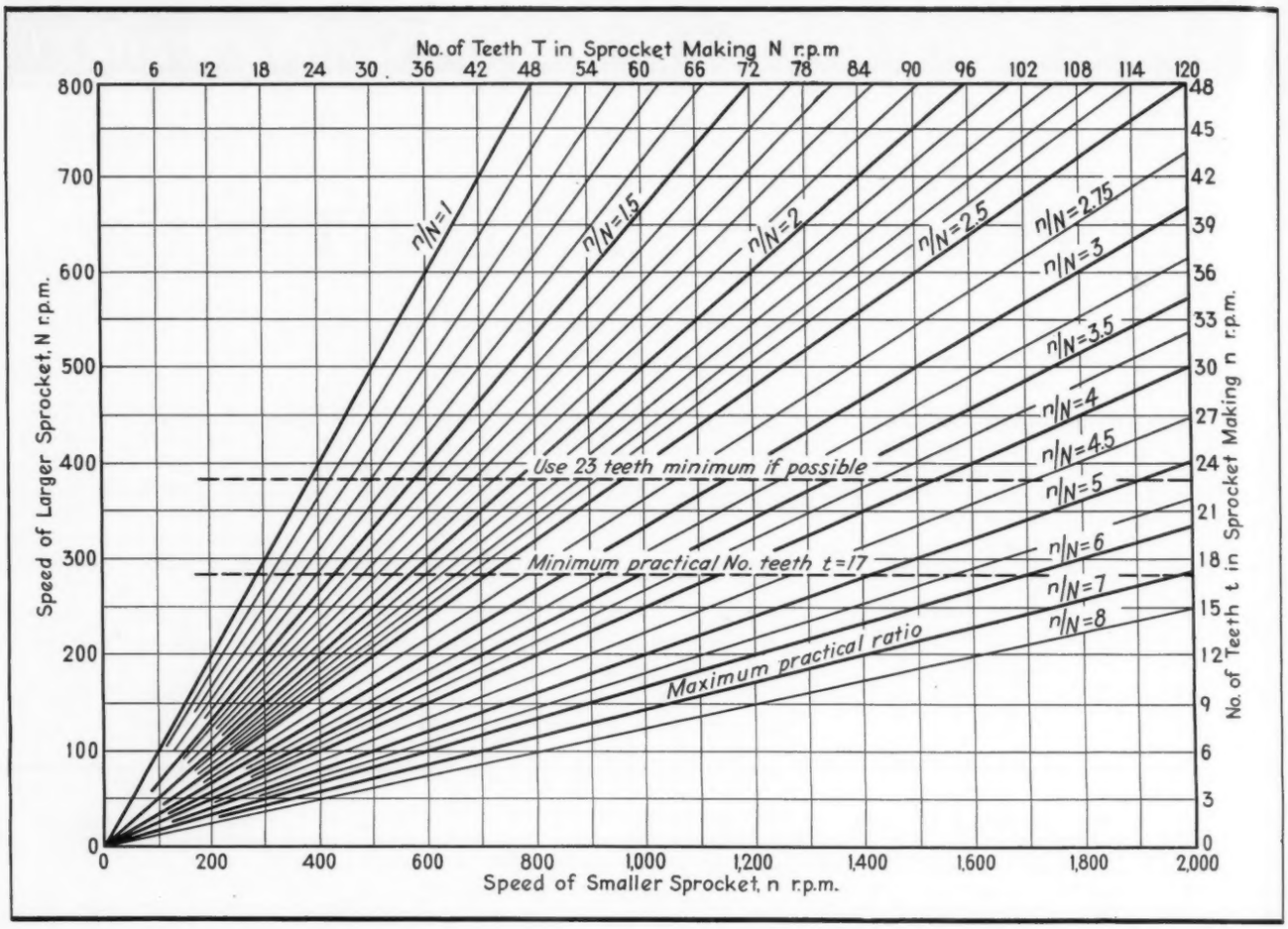

No. of Teeth T in Sprocket Making N r.p.m
0 6 12 18 24 30 36 42 48 54 60 66 72 78 84 90 96 102 108 114 120
Speed of Larger Sprocket, N r.p.m.
800 700 600 500 400 300 200 100 0
No. of Teeth t in Sprocket Making n r.p.m.
48 45 42 39 36 33 30 27 24 21 18 15 12 9 6 3 0
n/N=1
n/N=1.5
n/N=2
n/N=2.5
n/N=2.75
n/N=3
n/N=3.5
n/N=4
n/N=4.5
n/N=5
n/N=6
n/N=7
n/N=8
Use 23 teeth minimum if possible
Minimum practical No. teeth t=17
Maximum practical ratio
0 200 400 600 800 1,000 1,200 1,400 1,600 1,800 2,000
Speed of Smaller Sprocket, n r.p.m.

Fig. 2—Use this chart to select size of sprockets for given speed ratio

term $K\ (T-t)^2/C_p$. Values for constant K in this previous equation have been worked out only to five decimals.

Since Equation (2) does not usually result in a chain which is a whole number of pitches in length, it is usually necessary to determine the exact center distance required for a chain whose length L is the whole number of pitches next greater than L_p. To do this, apply the following formula, where C_1 is the approximate center distance in inches as assumed when applying Equation (2), and P = pitch in in.

$$C = C_1 + P\,\frac{(L-L_p)}{2} \qquad (3)$$

Where greater accuracy is required, this process can be repeated.

Derivation of Formula

Let L = chain length in inches and C = center-to-center distance in inches. Then from Fig. 3,

$$L = \pi\,(R+r) + 2\theta\,(R-r) + 2\,C\cos\theta \qquad (a)$$

$$\text{also}\quad \sin\theta = \frac{R-r}{c} = X \qquad (b)$$

$$\theta = \sin^{-1} X$$

$$\text{then}\quad 2\,C\cos\theta = 2\,C\sqrt{1-X^2}\ \text{and} \qquad (c)$$

$$L = \pi\,(R+r) + 2\sin^{-1}X\,(R-r) + 2\,C\sqrt{1-X^2} \qquad (d)$$

from (b), $R-r = CX$, and (d) becomes, (e)

$$L = \pi\,(R+r) + 2CX\sin^{-1}X + 2\,C\sqrt{1-X^2} \qquad (f)$$

$$= \pi\,(R+r) + 2C\,[X\sin^{-1}X + \sqrt{1-X^2}] \qquad (g)$$

and $[X\sin^{-1}X + \sqrt{1-X^2}]$ can be expanded by Taylor's theorem:

$$[\quad] = 1 + \frac{X^2}{2} + \frac{1}{2\times3}\,\frac{X^4}{4} + \frac{1\times3}{2\times4\times5}\,\frac{X^6}{6} \quad . \quad . \quad . \qquad (h)$$

(Continued on page 336)

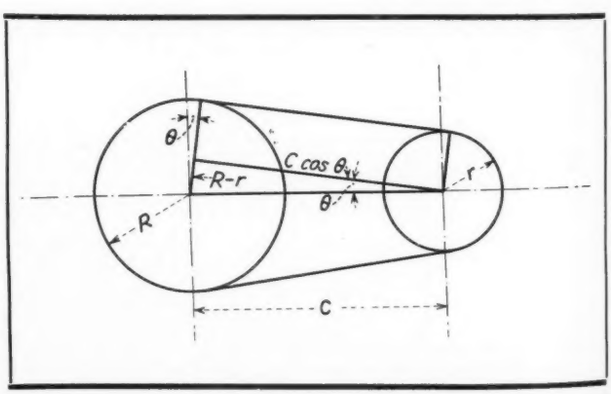


Fig. 3—Diagram of typical chain drive

Values of K to Apply in Chain Length Equation (2)

$\frac{T-t}{C_P}$	K	$\frac{T-t}{C_P}$	K	$\frac{T-t}{C_P}$	K	$\frac{T-t}{C_P}$	K	$\frac{T-t}{C_P}$	K	$\frac{T-t}{C_P}$	K
0.02	0.000507	1.02	0.025894	2.02	0.051622	3.02	0.078085	4.02	0.105830	5.02	0.135804
0.04	0.001013	1.04	0.026404	2.04	0.052143	3.04	0.078625	4.04	0.106403	5.04	0.136438
0.06	0.001520	1.06	0.026914	2.06	0.052664	3.06	0.079166	4.06	0.106977	5.06	0.137073
0.08	0.002026	1.08	0.027423	2.08	0.053185	3.08	0.079707	4.08	0.107552	5.08	0.137709
0.10	0.002533	1.10	0.027934	2.10	0.053707	3.10	0.080248	4.10	0.108128	5.10	0.138349
0.12	0.003040	1.12	0.028445	2.12	0.054229	3.12	0.080790	4.12	0.108705	5.12	0.138990
0.14	0.003546	1.14	0.028956	2.14	0.054751	3.14	0.081333	4.14	0.109282	5.14	0.139632
0.16	0.004053	1.16	0.029467	2.16	0.055273	3.16	0.081876	4.16	0.109861	5.16	0.140276
0.18	0.004559	1.18	0.029978	2.18	0.055796	3.18	0.082420	4.18	0.110441	5.18	0.140922
0.20	0.005066	1.20	0.030489	2.20	0.056319	3.20	0.082964	4.20	0.111021	5.20	0.141570
0.22	0.005574	1.22	0.031001	2.22	0.056842	3.22	0.083509	4.22	0.111603	5.22	0.142220
0.24	0.006080	1.24	0.031513	2.24	0.057366	3.24	0.084054	4.24	0.112185	5.24	0.142872
0.26	0.006587	1.26	0.032025	2.26	0.057890	3.26	0.084600	4.26	0.112769	5.26	0.143526
0.28	0.007093	1.28	0.032537	2.28	0.058414	3.28	0.085147	4.28	0.113353	5.28	0.144183
0.30	0.007600	1.30	0.033048	2.30	0.058939	3.30	0.085694	4.30	0.113939	5.30	0.144841
0.32	0.008107	1.32	0.033560	2.32	0.059464	3.32	0.086242	4.32	0.114526	5.32	0.145501
0.34	0.008614	1.34	0.034073	2.34	0.059989	3.34	0.086790	4.34	0.115113	5.34	0.146164
0.36	0.009121	1.36	0.034585	2.36	0.060515	3.36	0.087338	4.36	0.115702	5.36	0.146828
0.38	0.009628	1.38	0.035098	2.38	0.061041	3.38	0.087888	4.38	0.116291	5.38	0.147495
0.40	0.010135	1.40	0.035611	2.40	0.061567	3.40	0.088438	4.40	0.116882	5.40	0.148164
0.42	0.010642	1.42	0.036124	2.42	0.062104	3.42	0.088989	4.42	0.117474	5.42	0.148835
0.44	0.011149	1.44	0.036637	2.44	0.062621	3.44	0.089440	4.44	0.118066	5.44	0.149510
0.46	0.011657	1.46	0.037151	2.46	0.063148	3.46	0.089992	4.46	0.118660	5.46	0.150286
0.48	0.012164	1.48	0.037665	2.48	0.063676	3.48	0.090645	4.48	0.119256	5.48	0.150866
0.50	0.012671	1.50	0.038179	2.50	0.064204	3.50	0.091198	4.50	0.119852	5.50	0.151547
0.52	0.013179	1.52	0.038693	2.52	0.064733	3.52	0.091752	4.52	0.120449	5.52	0.152232
0.54	0.013687	1.54	0.039207	2.54	0.065262	3.54	0.092207	4.54	0.121048	5.54	0.152919
0.56	0.014194	1.56	0.039722	2.56	0.065791	3.56	0.092862	4.56	0.121647	5.56	0.153608
0.58	0.014702	1.58	0.040237	2.58	0.066321	3.58	0.093418	4.58	0.122247	5.58	0.154301
0.60	0.015209	1.60	0.040752	2.60	0.066851	3.60	0.093974	4.60	0.122849	5.60	0.154996
0.62	0.015717	1.62	0.041267	2.62	0.067382	3.62	0.094531	4.62	0.123453	5.62	0.155694
0.64	0.016225	1.64	0.041782	2.64	0.067913	3.64	0.095089	4.64	0.124058	5.64	0.156397
0.66	0.016733	1.66	0.042298	2.66	0.068444	3.66	0.095648	4.66	0.124664	5.66	0.157101
0.68	0.017241	1.68	0.042813	2.68	0.068976	3.68	0.096207	4.68	0.125271	5.68	0.157809
0.70	0.017749	1.70	0.043329	2.70	0.069509	3.70	0.096767	4.70	0.125879	5.70	0.158521
0.72	0.018258	1.72	0.043846	2.72	0.070041	3.72	0.097328	4.72	0.126489	5.72	0.159236
0.74	0.018766	1.74	0.044362	2.74	0.070574	3.74	0.097889	4.74	0.127100	5.74	0.159954
0.76	0.019274	1.76	0.044879	2.76	0.071108	3.76	0.098451	4.76	0.127713	5.76	0.160676
0.78	0.019782	1.78	0.045395	2.78	0.071642	3.78	0.099014	4.78	0.128328	5.78	0.161404
0.80	0.020291	1.80	0.045913	2.80	0.072176	3.80	0.099578	4.80	0.128943	5.80	0.162131
0.82	0.020800	1.82	0.046431	2.82	0.072711	3.82	0.100142	4.82	0.129558	5.82	0.162865
0.84	0.021309	1.84	0.046949	2.84	0.073246	3.84	0.100707	4.84	0.130176	5.84	0.163604
0.86	0.021818	1.86	0.047467	2.86	0.073782	3.86	0.101273	4.86	0.130795	5.86	0.164346
0.88	0.022327	1.88	0.047985	2.88	0.074318	3.88	0.101840	4.88	0.131416	5.88	0.165093
0.90	0.022836	1.90	0.048504	2.90	0.074855	3.90	0.102408	4.90	0.132039	5.90	0.165845
0.92	0.023345	1.92	0.049023	2.92	0.075392	3.92	0.102976	4.92	0.132662	5.92	0.166601
0.94	0.023855	1.94	0.049542	2.94	0.075930	3.94	0.103545	4.94	0.133288	5.94	0.167362
0.96	0.024364	1.96	0.050062	2.96	0.076468	3.96	0.104115	4.96	0.133915	5.96	0.168128
0.98	0.024874	1.98	0.050581	2.98	0.077006	3.98	0.104685	4.98	0.134543	5.98	0.168901
1.00	0.025384	2.00	0.051102	3.00	0.077545	4.00	0.105256	5.00	0.135172	6.00	0.169680

and $L = \pi\,(R+r) + 2C + 2C\,X\left[\frac{X}{2} + \frac{1}{2\times3}\,\frac{X^3}{4} + \frac{1\times3}{2\times4\times5}\,\frac{X^5}{6}\right]$

$= \pi\,(R+r) + 2C + 2C\;X\;S$ (i)

where S = sum of series.

Equating the last two terms of (g) and (i),

$2C\,[X \sin^{-1} X + \sqrt{1-X^2}] = 2C\;(1 + XS)$

and $S = \sin^{-1} X + \frac{\sqrt{1-X^2}}{X} - \frac{1}{X} = \sin^{-1} X - \frac{1-\sqrt{1-X^2}}{X}$ (j)

Now let

T = number teeth in large wheel
t = number teeth in small wheel
p = chain pitch in in.

then $2\pi R = pT$ $\qquad$ $2\pi r = pt$

$R = \frac{p}{2\pi}\,T,$ $\qquad$ $r = \frac{p}{2\pi}\,t$

and $R - r = \frac{p}{2\pi}\,(T-t)$ $\qquad$ $R + r = \frac{p}{2\pi}\,(T+t)$

$X = \sin\theta = \frac{R-r}{C} = \frac{p}{2\pi}\,\frac{(T-t)}{C}$ (k)

Substituting above values in equation (i),

$L = \frac{p\,(T+t)}{2} + 2C + 2C\,\frac{p}{2\pi}\;\frac{T-t}{C}\,S$

let $L/P = L_p$ = chain length in pitches, then

$L_p = \frac{T+t}{2} + \frac{2C}{p} + \frac{S}{\pi}\,(T-t)$ (l)

let $\frac{S}{\pi} = K$, and $C/p = C_p$ = center distance in pitches

then $L_p = \frac{T+t}{2} + 2\,C_p + K\,(T-t)$ (m)

Evaluation of Constant K

$K = \frac{S}{\pi} = \frac{1}{\pi}\left[\sin^{-1} X - \frac{1-\sqrt{1-X^2}}{X}\right]$ (n)

The table headings are in terms of $\frac{T-t}{C} = 2\pi\sin\theta$, so

let $\frac{T-t}{C} = d$, then $X = \sin\theta = \frac{d}{2\pi}$ and substituting in (n),

$K = \frac{1}{\pi}\left[\sin^{-1}\frac{d}{2\pi} - \frac{1}{d}\,(2\pi - \sqrt{4\pi^2 - d^2})\right]$ (o)

The above was used to compute values of K at d = 1, 2, 3, 4, 5, and 6, using 6-place tables.

Evaluation by series of K in interval $d = 0$ to $d = 1$:

$K = S/\pi$, and substituting $X = d/2\pi$ in (i) and (n)

$K = \frac{1}{2\pi}\left[\frac{d}{2\pi} + \frac{1}{3\times4}\left(\frac{d}{2\pi}\right)^3 + \frac{1\times3}{4\times5\times6}\left(\frac{d}{2\pi}\right)^5 + \frac{1\times3\times5}{4\times6\times7\times8}\left(\frac{d}{2\pi}\right)^7 \cdots\right]$

$K = \frac{d}{4\pi^2} + \frac{d^3}{192\pi^4} + \frac{d^5}{2560\pi^6} + \frac{5}{114688\pi^8} \cdots$

$K = 0.025330\,d + 0.000053469\,d^3 + 0.0000004063\,d^5 + 0.00000000459\,d^7$ (p)

The above series was nice to use up to $d = 1$ but above this some other series had to be found.

Substitute $X = \frac{d}{2\pi}$ in equation (o)

then $K = \frac{1}{\pi}\left[\sin^{-1}\frac{d}{2\pi} - \frac{1}{d}\,(2\pi - \sqrt{4\pi^2 - d^2}\,)\right]$ (q)

now expanding function about $(d - h)$

where let us put $m = \sqrt{4\pi^2 - d^2}$

then $K = \frac{1}{\pi}\left\{\sin^{-1}\frac{h}{2\pi} - \frac{1}{h}\,(2\pi - m^{1/2}) + \frac{1}{h}\,(2\pi - m^{1/2})\,\frac{d-h}{h}\right.$

$+\left[-\frac{1}{h}\,(2\pi - m^{1/2}) + \frac{h}{2m^{1/2}}\right]\frac{(d-h)^2}{h^2}$

$+\left[\frac{1}{h}\,(2\pi - m^{1/2}) - \frac{h}{2m^{1/2}} - \frac{h^3}{6m^{3/2}}\right]\frac{(d-h)^3}{h^3}$

$\left.+\left[-\frac{1}{h}\,(2\pi - m^{1/2}) + \frac{h}{2m^{1/2}} + \frac{h^3}{8m^{5/2}}\right]\frac{(d-h)^4}{h^4} \cdots\right\}$

The result of the expansion is given above, but all intermediate steps for getting the derivatives of K have been omitted.

Only three terms of the above expansion were used and h is close enough to d so that results are accurate to the sixth decimal place. The K computed by (o), and the series in $(d - h)$ check for largest value of $(d - h)$ used.

After tabulating the results of computations, the difference between successive values of K were found. Since increments of d were taken as 0.02 throughout, the K differences increase uniformly from $d = 0.02$ up to $d = 6.00$.

Example of Methods

A chain drive has a large sprocket with 55 teeth, a small sprocket with 17 teeth and a center-to-center distance of 24 in. Pitch of the chain is ¾ in. First make a rapid estimation, then an accurate determination of chain length.

Estimation—applying Equation (1),

$L_p = \frac{55+17}{2} + \frac{2\times24}{3/4} + 1.2 = 101.2$ pitches

The last term, A, in the above equation was found by entering the chart, Fig. 1, at $C_p = 32$, moving horizontally to the diagonal line $(T - t) = 38$, and then moving vertically from this intersection to read $A = 1.2$ pitches at bottom of chart.

Accurate determination—The first two terms in Equation (2) are the same as for Equation (1). For the last term compute $(T - t)/C_p = 1.20$. Then referring to the table of values for K, find $K = 0.030489$ for $(T - t)/C_p = 1.20$. Then the formula becomes:

$L_p = \frac{55+17}{2} + \frac{2\times24}{3/4} + 0.030489\,(55 - 17) = 101.158$ pitches

To find the exact center distance for a chain with the nearest greater whole number of pitches in length (in this case 102 pitches), apply Equation (3);

$C = 24 + 3/4\left(\frac{102 - 101.158}{2}\right) = 24.315$ in.

This value, when checked back with Equation (2), requires an accurate chain length of 101.979 pitches.